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OPEN FIRES

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F O R E W O R D

This research program has been sponsored by the Forest Service of the United States Department of Agriculture under Grants FG-Sp-114 and FG-Sp-146, monitored by the European Regional Research Office in Rome of the Agricultural Research Service. The research work has been performed at the Propulsion Department of the Instituto Nacional de Técnica Aeroespacial Esteban Terradas. Madrid. Spain. The complete research program of both Grants comprised two different problems: The study of the basic laws controlling open fires and the study of the combustion properties and flight paths of burning embers of wood (firebrands). According to this the final Report of the Grants has been divided in two volumes: Open Fires, and Transport and Combustion of Firebrands.

A C K N O W L E D G E M E N T

It is aknowledged the valuable advice given by Professor A.Liñán and the valuable partial collaborations given by Mr. J.A. Bollain and Mr. R. Elvira Pisón. The authors are also indebted to the work performed by the laboratory assistants C.García Fernández, M.Muñoz and F.Silva; draftsman E.Goñi and calculists J.M.Valero and R.Almarza; photographer W.Pantoja and typist Miss.F.Moreda.

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A B S T R A C T

The research program has comprised the experimental and theoretical study of some basic laws of open fires. This study has been carried out by utilizing the pool fire technique, which consists in burning liquid fuels in cylindrical vessels, and in measuring and analyzing burning rates, energy balances, flame properties and fuel temperatures.

Several important conclusions have been obtained on burning rates, energy balances and flame characteristics including the influence of type of fuel, vessel size and vessel configuration.

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INTRODUCTION

The study of the fundamental laws controlling the process of an open fire is extremely difficult.

The combustion process in a large fire is a turbulent flame resulting from the evaporation of solid or liquid fuels and mixing of these vapors with the air. This evaporation takes place by means of the heat transmitted from the flame through convection and radiation.

On the other hand, the aerodynamic field is very complicated since it is controlled by a free-convection process in which viscosity may exert an important influence.

The over-all phenomenon results from the interaction between the combustion process and the aerodynamic field, which originates a process of an extremely complicated nature.

A number of studies on the combustion process ignoring the aerodynamic field have been carried out by several investigators such as those of Refs. 1-9.

Most of these studies are only experimental and the theoretical treatments have been generally restricted to derivations of semi-empyrical expressions for the burning rates.

On the other hand, some studies of the aerodynamic of fires, especially in connection with convection plumes have been considered, as those of Refs. 10-13.

In order to clarify some aspects of the problem of

Open Fires, a research group of the Instituto Nacional de Técnica Aeroespacial of Madrid has conducted a research program under Grants Fg-Sp-114 and Fg-Sp-146 sponsored by the Forest Service of the U.S. Department of Agriculture.

The research program has consisted in the study of axisymmetrical fires produced by burning liquid fuels contained in cylindrical vessels (burners) in the open atmosphere.

The investigation has been directed towards the study of the basic laws governing the process, which relate the physico-chemical properties of the fuel with the observable characteristics of a fire, such as flame temperature, size and emissivity of the flame, burning rate, etc.

The study has been conducted by carrying out a series of measurements and by studying theoretically some partial processes which are part of the over-all process.

A model of the process of the pool fire has been developed, which gives the burning rates and temperature profiles within the fuel as functions of the amount of heat received by the fuel from the flame.

This model gives separately the influence of both radiation and convection heat transfer mechanisms, taking into account fuel properties and geometrical configuration of pool and fire. Transient state has also been considered which exists after ignition and before the fire attains quasi-stationary combustion conditions.

Several conclusions, which are listed at the end of this work, have been obtained which may clarify some aspects of the very complex process of open fires.

I. THEORETICAL STUDIES

1. DESCRIPTION OF THE POOL FIRE

From direct observations of pool fires it has been concluded that the flames have an inner core of fuel vapor immediately above the liquid fuel surface which has approximately a conical shape. The temperature within that zone is small as compared to the flame temperature and its absorption coefficient is practically negligeable. (Fig.1)

The inner core exists in both laminar and turbulent flames. In the laminar flame produced by a small burner, the inner core is large and ratio of its height $z_{v,max}$ to burner diameter D may reach considerable values.

On the other hand, for large burners the flame is turbulent, the air is entrained towards the center of the burner and ratio $z_{v,max}/D$ reduces considerably.

The situation changes considerably for the case of the flame produced by a gas jet. In this case the fuel is forced towards the combustion region and that inner core, which also exists, is very large.

Temperature profiles across a turbulent flame may be approximated by means of "top hat" profiles, since temperature changes abruptly in the boundaries of the luminous region and it does not change very much within the flame. Furthermore, in the longitudinal direction the temperature of a turbulent flame changes little also. Therefore, a turbulent flame will be considered as a volume of gas at uniform temperature.

On the other hand, in laminar flames sharp temperature profiles exist. However the assumption of uniform temperature within the flame is only important from the radiation point of view and in laminar flames radiation is of very little importance.

The combustion process can be divided into the following periods:

a) Heating and ignition. The fuel is heated from an external source (a flame) until its surface reaches the flash point temperature, and then, combustion begins.

b) Combustion and surface heating. When combustion starts the fuel surface temperature increases very rapidly until it reaches the boiling temperature.

c) Transient combustion at constant fuel surface temperature. Combustion proceeds at constant fuel surface temperature. Temperature within the fuel increases and the burning rate also increases both tending towards stationary values.

The time required for periods a) and b) is very short as compared with the time of period c) and is function of the external mechanism used for ignition and of the burner configuration. For simplicity, periods a) and b) can be joined in only one period, which will be called heating and ignition, in which the fuel is heated until its surface reaches the boiling temperature. Burning rate (fuel consumption) can be disregarded throughout this process.

d) Stationary combustion. After a certain time combustion reaches, practically, stationary conditions, which can be directly calculated.

When using constant level cylindrical vessels for studying open fires in order to keep the fuel at a constant level one or several overflow tubes placed within the vessel are used. In this case, all fuel flow is heated up to the boiling temperature but a certain percentage of it is returned from the surface down through the overflow tube. We will see that this return flow influences considerably the values of the burning rates.

2. MODEL OF THE POOL FIRE

For the study of open fires using the pool-burner technique several types of these pool-burner models can be used regardless of their size. Considering the basic aspect of this study pool-burners with over-flow have been selected. This allows the utilization of an additional variable which is the amount of hot fuel flow returned from the vessel, variable which is easily controlled. (Fig.1)

In order to estimate the over-all thermal balance of a fire and the distribution of heat transfer between convective and radiant heat, a model of the fire has been developed based upon the following general assumptions:

a) The fuel enters the vessel through its bottom at ambient temperature T_o .This temperature is kept uniform and constant throughout the bottom by means of water cooling at T_o.The fuel moves upwards till it reaches the surface where part of the

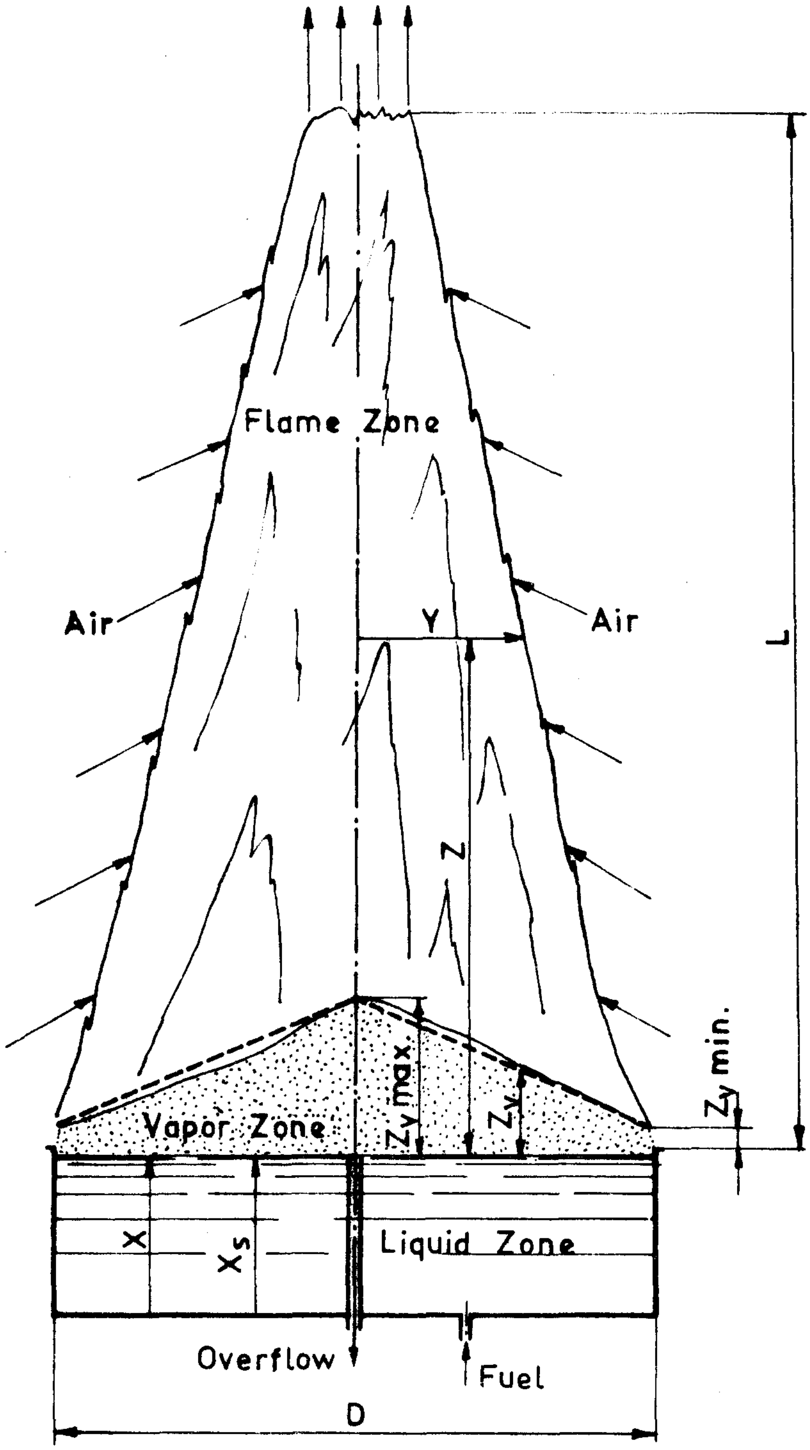


Fig.1. Pool Fire Model

fuel is evaporated and burned and the rest of the fuel is removed from the fuel surface without burning by means of overflow tubes. (Fig.1).

b) One-dimensional conditions will be considered by assuming that vessel diameter is large as compared to vessel depth and by assuming that conditions at the fuel surface are uniform.

c) Viscosity and gravitational forces are not considered. Density, thermal conductivity and specific heat of the fuel are taken as constant regardless of fuel temperature.

d) Temperature at the fuel surface will be taken equal to the boiling temperature at ambient pressure.

e) The flame (luminous region) is assumed to be constituted by a volume of gases of uniform temperature.

In order to facilitate the study the following zones have been separately studied: liquid fuel zone, vapor fuel zone, and luminous zone (flame).

3. HEAT TRANSFER IN THE LIQUID ZONE

3.1 General Equations

Under the aforementioned assumptions the equation of energy within the fuel, neglecting the kinetic energy, expresses the transport balance of heat and internal energy:

$$v \frac{\partial T}{\partial x} + \frac{\partial T}{\partial t} = \frac{\lambda}{\rho c} \frac{\partial^2 T}{\partial x^2} \qquad (1)$$

The heat received at the fuel surface is

$$\dot{q}_s = \dot{m}_b q_l + \dot{q}_{cl} = \dot{m}_b q_l + \lambda \left(\frac{dT}{dx}\right)_s \qquad (2)$$

in which q_l is the latent heat of evaporation, $\dot{m}_b$ is the fuel flow evaporated and burned, and $\dot{q}_{cl}$ the heat conducted into the liquid.

Introducing the dimensionless variables and parameters:

$$\theta = \frac{T}{T_s} \qquad (3) \qquad\qquad \delta = \frac{x}{x_s} \qquad (4)$$

$$\tau = \frac{\lambda}{\rho c} \frac{t}{x_s^2} \qquad (5) \qquad\qquad \kappa_s = \frac{\dot{q}_s x_s}{\lambda T_s} \qquad (6)$$

$$\kappa_l = \frac{q_l}{T_s c} \qquad (7) \qquad\qquad \nu = \frac{\rho c}{\lambda} x_s v \qquad (8)$$

$$\frac{\dot{m}_b}{\dot{m}} = \xi \qquad (9)$$

Eqs. (1) and (2) are expressed, respectively, as follows:

$$\nu \frac{\partial \theta}{\partial \delta} + \frac{\partial \theta}{\partial \tau} = \frac{\partial^2 \theta}{\partial \delta^2} \qquad (10)$$

$$\kappa_s = \kappa_1 \xi \nu + \left(\frac{\partial \theta}{\partial \delta}\right)_{\delta=1} \tag{11}$$

3.2 Solution of the Equation. Stationary process.

The Eq. (10) governing the process is reduced to

$$\nu \frac{\partial \theta}{\partial \delta} = \frac{\partial^2 \theta}{\partial \delta^2} \tag{12}$$

with the following boundary conditions

$$\begin{aligned} \theta &= \theta_o \qquad \text{at} \qquad \delta = 0 \\ \theta &= 1 \qquad \text{at} \qquad \delta = 1 \end{aligned} \tag{13}$$

Eq.(12) is immediately integrated, resulting:

$$\frac{1 - \theta}{1 - \theta_o} = \frac{e^{\nu} - e^{\nu\delta}}{e^{\nu} - 1} \tag{14}$$

and the Eq.(11) can be written in the form:

$$\frac{\kappa_s}{\kappa_1} = \xi\nu + \frac{(1 - \theta_o)}{\kappa_1} \; \frac{\nu e^{\nu}}{e^{\nu} - 1} \tag{15}$$

In Fig.2, dimensionless temperature profiles are shown for different values of parameter ν . It may be seen that when the mass flow parameter ν is large, the temperature profiles are very sharp in the vicinity of the fuel surface.

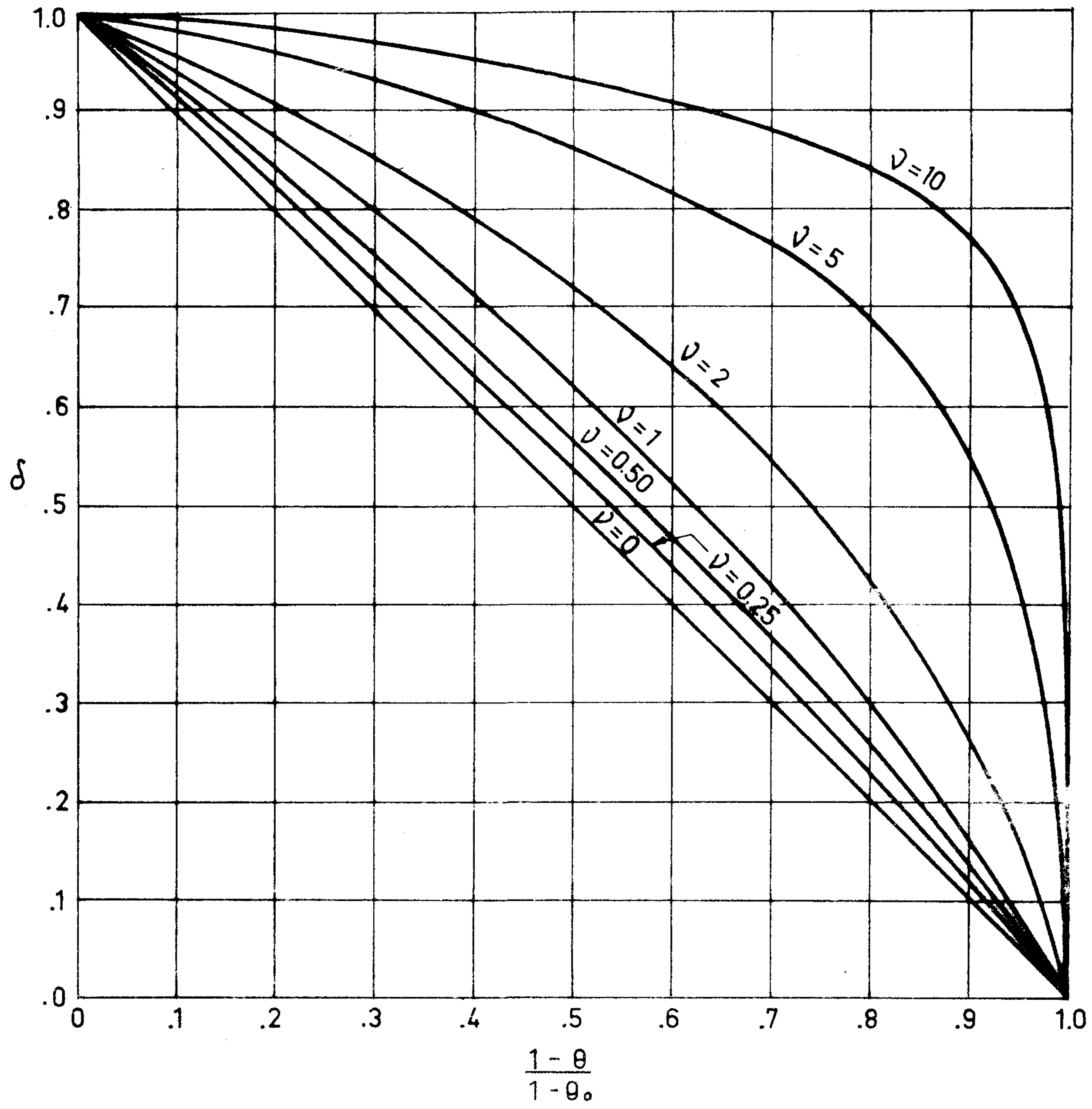


Fig.2. Dimensionless Temperature Profiles

Therefore, a thermocouple placed at the fuel surface would measure an average value of the temperature. This might explain why the measured values of the temperature at the fuel surface are usually somewhat smaller than the boiling temperature, which is the value predicted by theory.

In Fig.3, dimensionless heat conducted into the liquid, $\frac{\kappa_s}{\kappa_l} - \xi\nu$, is shown for different values of parameter $\frac{1-\theta_o}{\kappa_l}$, (Eq.15). This figure shows also the total heat needed for the evaporation of all the liquid mass flow, $\left(\frac{\kappa_s}{\kappa_l} - \xi\nu + \nu\right)$. For example, in the case of $\nu = 4$, $\frac{1-\theta_o}{\kappa_l} = 0{,}5$, the magnitude AB represents the heat conducted through the liquid and BC represents the evaporation heat. For a given value of the heat transmitted to the liquid surface, and a given value of the parameter $\frac{1-\theta_o}{\kappa_l}$, for example: $\frac{\kappa_s}{\kappa_l} = 6$, $\frac{1-\theta_o}{\kappa_l} = 0{,}5$, the segment BC gives the mass flow burned without overflow, and the segment DE gives the mass flow burned with overflow; $DE = \xi\nu = 3{,}5$, $\nu = 5$. From the figure, it can be deduced the very important influence of the overflow on the burning rate.

When $\nu >> 1$ the Eq.(15) gives

$$\dot{q}_s = \dot{m}\left[\xi q_l + c\,(T_s - T_o)\right] \qquad (16)$$

3.3 Solution of the Equation . Transient Process

Eq.(10) governs the transient process. The initial conditions of this transient process can be very different since they depend upon the ignition procedure and they also

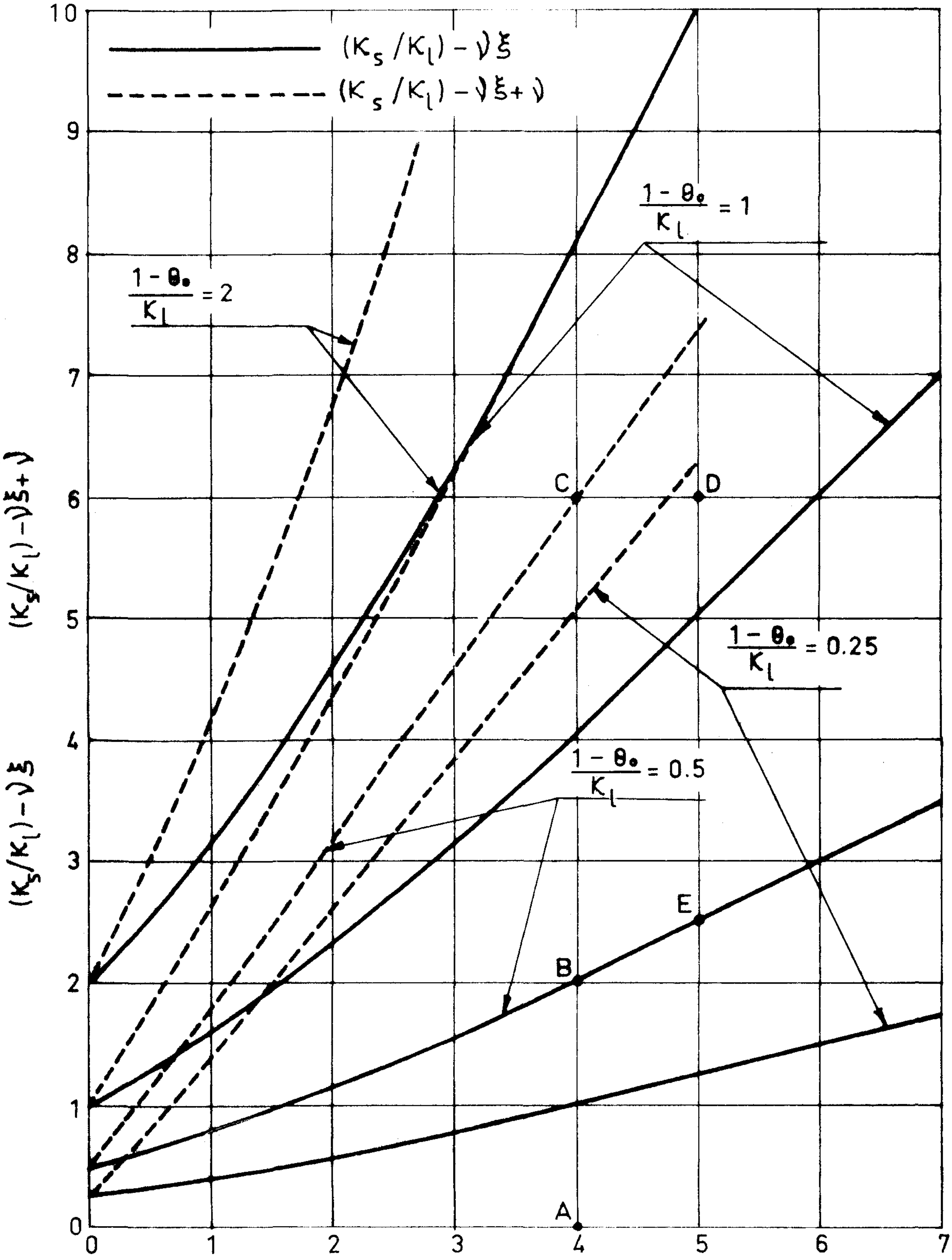


Fig.3. Dimensionless Heat Conducted into the Liquid

depend on the existence or absence of the overflow system.

A normal ignition procedure consists in approaching a pilot flame to the fuel surface, which is followed by spreading of the flame over the entire surface in a very short time when initial temperature of the fuel is higher than flash temperature.

Initially the flame is located very close to the fuel surface and therefore it can be assumed that boiling temperature at the fuel surface is reached in a very short time. Solution of Eq.(10) is simplified by the existence of an initial overflow higher than the mass flow burned, because in this case ν is constant. In the transient process not only the fuel conditions depend on time but the heat transferred from the flame to the fuel surface also, since the size of this flame changes.

The complete study of the transient process is very complex and it is of secondary importance considering the actual estate of knowledge on open fires.

However, some conclusions may be obtained by analyzing the process in the liquid region. According to the aforewritten hypothesis boundary conditions of the selected model are as follows:

$$\begin{array}{llll} \theta = \theta_o & \text{at} & \tau = 0 & \\ \theta = \theta_o & \text{at} & \delta = 0 \quad , & \tau \geq 0 \\ \theta = 1 & \text{at} & \delta = 1 \quad , & \tau > 0 \end{array} \qquad (17)$$

Solution of Eq.(10) with boundary conditions (17), taking into account that ν is constant, is

$$\frac{1-\theta}{1-\theta_o} = \frac{e^{\nu}-e^{\nu\delta}}{e^{\nu}-1} - e^{-\frac{\nu(1-\delta)}{2}} \sum_n (-1)^n \frac{8\pi n}{\nu^2+4\pi^2 n^2} e^{-\frac{(4\pi^2 n^2+\nu^2)\tau}{4}} \operatorname{sen}\pi n\delta \tag{18}$$

and Eq.(11) can be written in the form:

$$\frac{\kappa_s}{\kappa_1} = \nu\xi + \frac{1-\theta_o}{\kappa_1}\left[\frac{\nu e^{\nu}}{e^{\nu}-1} + \sum_n \frac{8\pi^2 n^2}{\nu^2+4\pi^2 n^2} e^{-\frac{(4\pi^2 n^2+\nu^2)\tau}{4}}\right] \tag{19}$$

In Fig.4 dimensionless temperature profiles $\theta = f(\delta,\tau)$ are shown for several values of ν .

Fig.5, shows dimensionless transient heat conducted through the liquid $\Gamma = \sum_n \frac{8\pi^2 n^2}{\nu^2+4\pi^2 n^2} \exp.\left[-\frac{(4\pi^2 n^2+\nu^2)\tau}{4}\right]$ as a function of time. From this figure and from Eq.(5) the influence of the thermal diffusivity of the liquid $\lambda/\rho c$ and depth vessel x_s on the transient period into the liquid zone can be analized. It may be seen the small influence of parameter ν on the transient heat conducted.

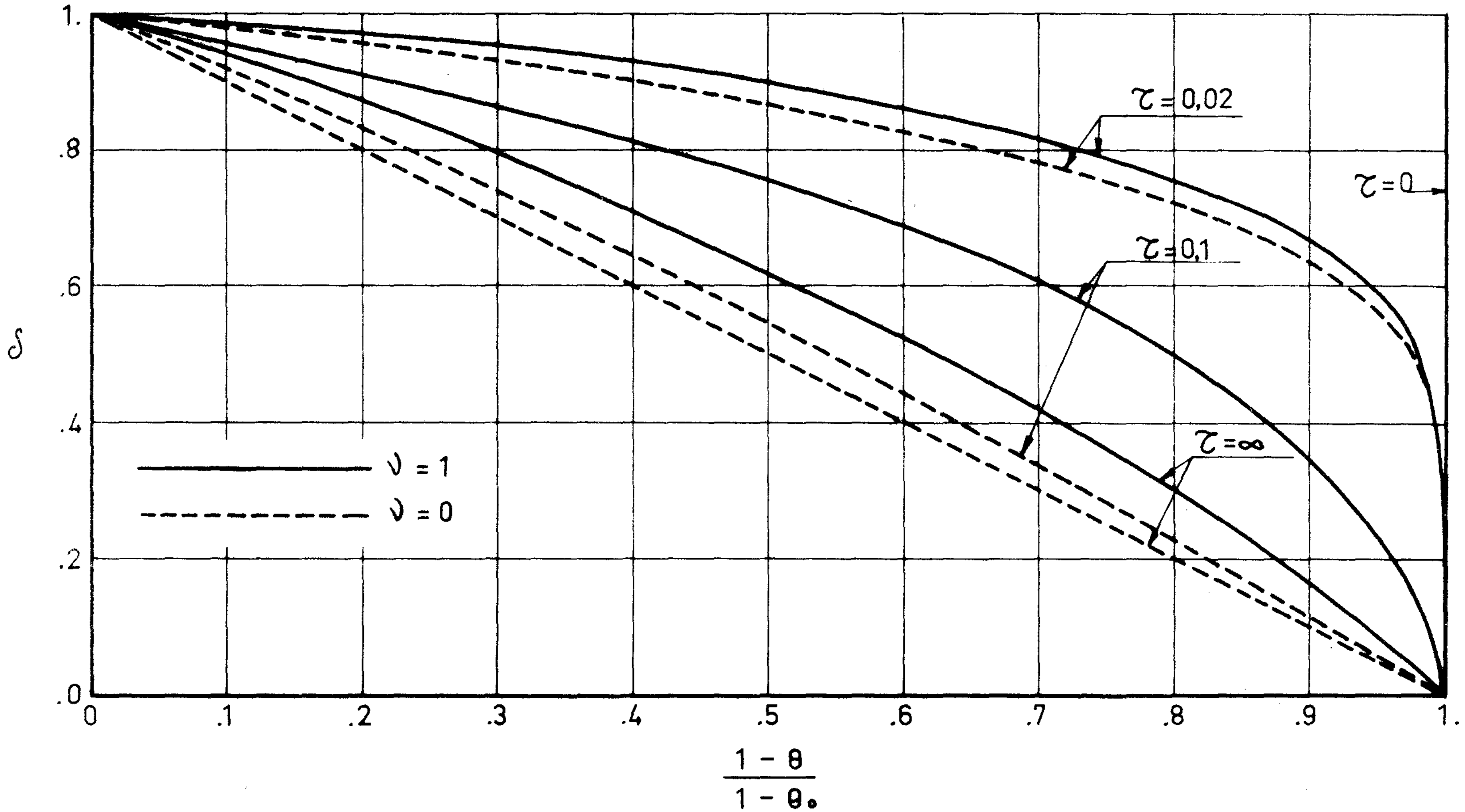


Fig.4. Transient Process. Dimensionless Temperature Profiles

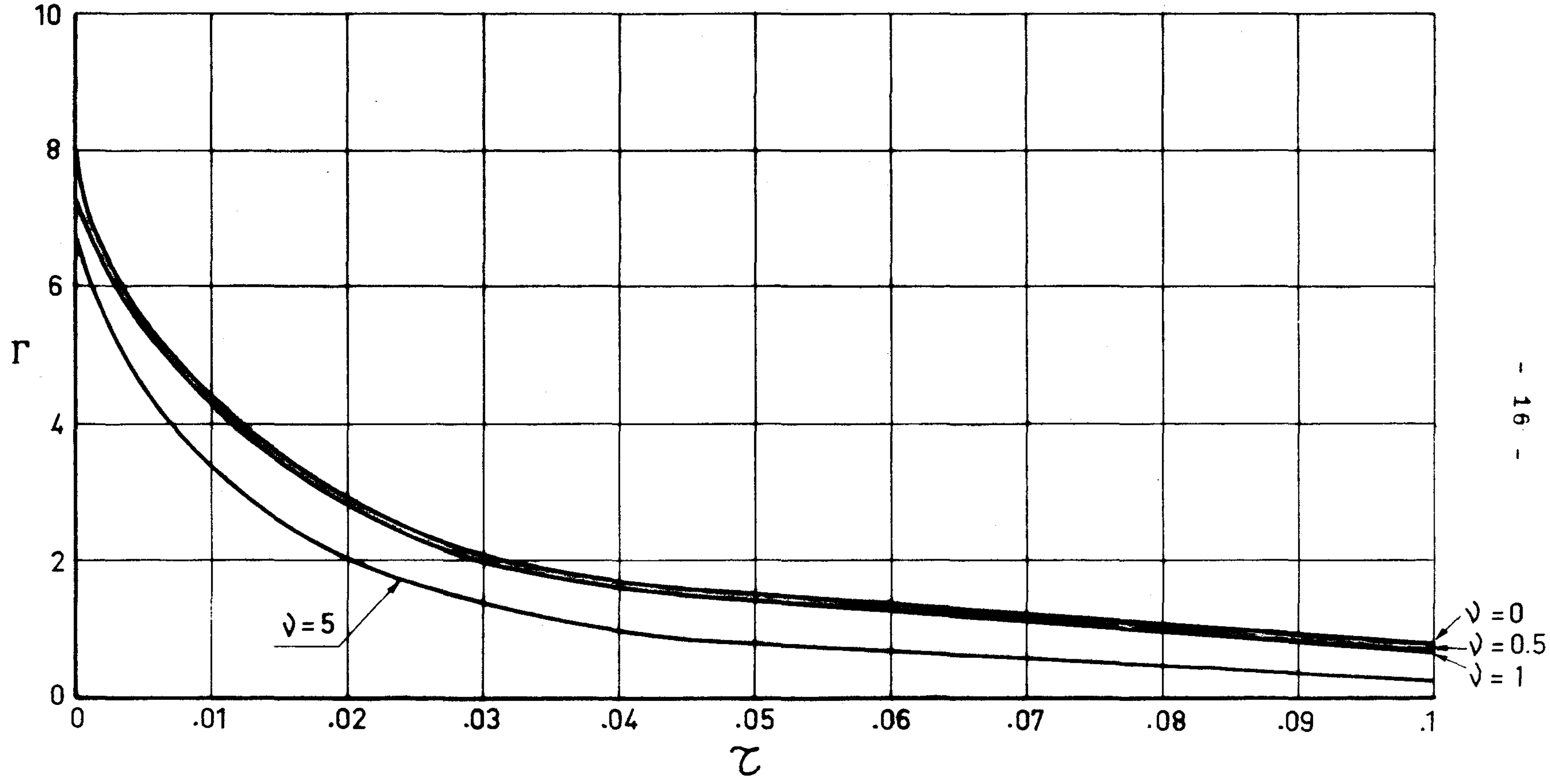


Fig.5. Transient Process. Dimensionless Heat Conducted into the Liquid.

4. HEAT TRANSFER IN THE VAPOR FUEL ZONE

4.1 General Equations

The value of the heat flow reaching the fuel surface per unit area through convection will be calculated by assuming that the motion of the fuel vapors is one-dimensional and that the heat transferred towards the OY direction is negligeable as compared to the heat transferred towards the OZ direction. (Fig.1)

The equation of energy which governs the heat transport phenomenon is Eq.(10) written in the form

$$\nu_v \frac{\partial \theta}{\partial \delta_v} + \frac{\partial \theta}{\partial \tau_v} = \frac{\partial^2 \theta}{\partial \delta_v^2} \qquad (20)$$

in which:

$$\nu_v = \frac{z_v \, c_v \, \dot{m}_b}{\lambda_v} \qquad (21) \qquad\qquad \delta_v = \frac{z}{z_v} \qquad (22)$$

$$\tau = \frac{\lambda_v \, t}{\rho_v \, c_v \, z_v^2} \qquad (23)$$

and where subscript v denotes fuel vapor.

4.2 Solution of the Equation . Stationary Process

Eq.(20) reduces to:

$$\nu_v \frac{\partial \theta}{\partial \delta_v} = \frac{\partial^2 \theta}{\partial \delta_v^2} \qquad (24)$$

and boundary conditions are as folllows:

$$\begin{aligned} \theta &= 1 \qquad \text{at} \qquad \delta_v = 0 \\ \theta &= \theta_f \qquad \text{at} \qquad \delta_v = 1 \end{aligned} \qquad (25)$$

The integration of Eq.(24) with boundary conditions (25) gives:

$$\frac{\theta - 1}{\theta_f - 1} = \frac{e^{\nu_v \delta_v} - 1}{e^{\nu_v} - 1} \qquad (26)$$

The heat $\dot{q}_{cv}$ received through convection per unit area by the fuel surface is:

$$\dot{q}_{cv} = \lambda_v \left(\frac{dT}{dz}\right)_{z=o} = \lambda_v \frac{T_s}{z_v} (\theta_f - 1) \frac{\nu_v}{e^{\nu_v} - 1} \qquad (27)$$

The heat $\dot{q}_{cv}$ depends on z_v , and, therefore, it varies along the burner radius. For the case shown in Fig.1, it results:

$$\frac{z_v}{z_{v,min}} = 1 + (\beta - 1) \left(1 - \frac{2r}{D}\right) \qquad (28)$$

in which $\beta = \dfrac{z_{v,max}}{z_{v,min}}$

The variation of $\dot{q}_{cv}$ with the vessel radius is not very significant for large burners. It is interesting to point out that the simplificatory assumption that $\dot{m}_b$ remains constant holds , for large vessels, because the radiant heat flow reaching the fuel surface decreases from the center to the vessel rim.

The mean value $\bar{\dot{q}}_{cv}$ of heat received per unit area taking account (27) and (28) is:

$$\bar{\dot{q}}_{cv} = \frac{4}{\pi D^2} \int_o^{D/2} \dot{q}_{cv} 2\pi r dr = \frac{T_s \lambda_v}{z_{v,min}} (\theta_f - 1)\ \phi_v \qquad (29)$$

in which:

$$\phi_v = \frac{2}{\beta - 1} \left[-2 \ln \left(1 - e^{-\nu_{v,min}}\right) - \right.$$

$$\left. - \frac{1}{\nu_{v,min}\ (\beta - 1)} \sum_{n=1}^{\infty} \frac{\left(1 - e^{\nu_{v,max}}\right)^n - \left(1 - e^{-\nu_{v,min}}\right)^n}{n^2} \right] \qquad (30)$$

The value of ϕ_v is shown in Fig.6. It may be seen in that figure how ϕ_v decreases rapidly as ν_v (that is to say $\dot{m}_b$ or $z_{v,min}$) increases. This result implies that the convective heat from the flame is mainly used in heating up the fuel vapors, but it does not practically reach the fuel surface.

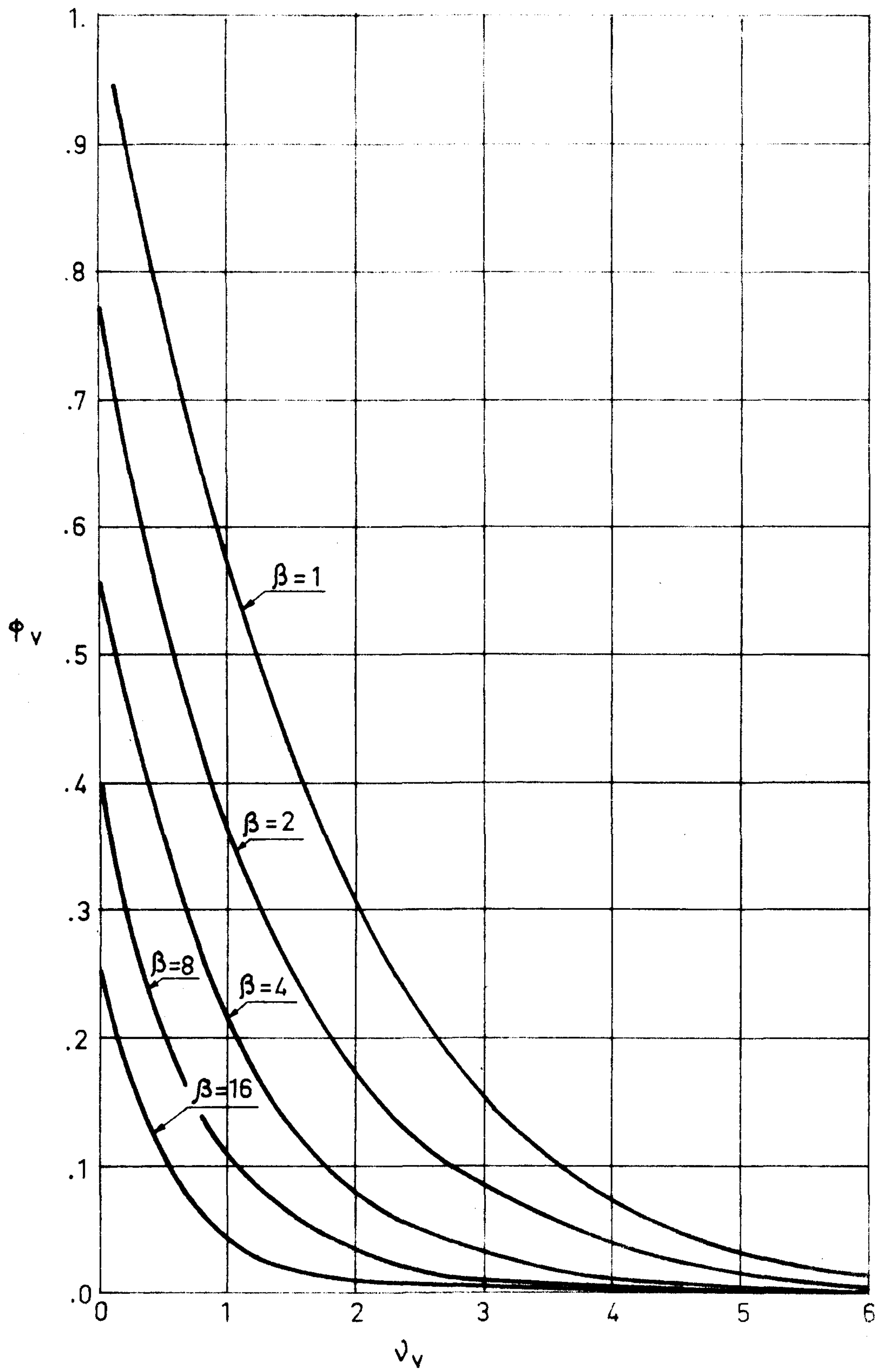


Fig.6. ϕ_v, Coefficient of Convective Heat from the **Flame**

4.3 Transient Process

The study of this process is similar to that of the liquid zone. But in this vapor zone, Eq.(5) for this case gives:

$$\frac{\lambda_v}{\rho_v \; c_v \; z_v^2} >> \frac{\lambda}{\rho c x_s^2} \qquad (31)$$

and therefore, the influence of this transient process is small as compared with the influence of the transient process in the liquid zone.

5. HEAT TRANSFERRED THROUGH RADIATION BY THE FLAME

5.1 Flame Emissivity

The radiant heat received by the fuel surface depends on the size, shape, temperature and on the emission coefficient of the flame.

The heat received by the liquid per unit area is given by:

$$\dot{q}_{rf} = \varepsilon_f \sigma T_f^4 - \varepsilon_l \sigma T_s^4 \approx \varepsilon_f \sigma T_s^4 \theta_f^4 \qquad (32)$$

since $T_s << T_f$.

The value of ε_f is not constant over the fuel surface. An average value of ε_f for the total surface may be estimated by assuming, as far as radiation is concerned,that the flame occupies not only its own volume, but also the volume of the inner core of fuel vapors, which is very small as compared with the volume of the flame, and by taking the equivalent radiant hemisphere approximation.

The value of flame temperature depends on the heat of combustion and on the air-flow entrained into the flame. An exact determination of this flame temperature is a difficult problem.

The average emissivity is, then, given by:

$$\bar{\varepsilon}_f = 1 - e^{-\alpha L_h} \qquad (33)$$

in which α is the absortion coefficient, and L_h is the radius of the equivalent radiant hemisphere or mean beam length. The value of α depends on the type of fuel, and especially on the existence of solid particles in the flame. It will be obtained experimentally. The value of L_h depends on the size and shape of the flame and it may be obtained by means of numerical integrations. However, a very approximated value of it is given by the expression:

$$L_h = 4 \frac{\text{Flame Volume}}{\text{Total Flame Surface}} = 4 \times \text{hydraulic radius} \qquad (34)$$

For the case of a conical flame, we obtain:

$$L_h = \frac{2}{3} \frac{D}{\frac{D}{2L} + \sqrt{1 + 4 \left(\frac{D}{L}\right)^2}} \qquad (35)$$

and for a cylindrical flame:

$$L_h = \frac{D}{\frac{D}{2L} + 1} \tag{36}$$

Mean beam length L_h may be approximately measured in a flame by means of the expression:

$$L_h = \frac{L}{1 + \left(\frac{L.D}{A_{fr}}\right)^2 + 4\,\frac{L^2}{A_{fr}}} \tag{37}$$

where A_{fr} is the cross-section area of the flame as seen in a lateral photograph.

Fig.7 shows the average emissivity $\bar{\varepsilon}_f$ for conical and cylindrical flames of several sizes and shapes. It may be observed how $\bar{\varepsilon}_f$ increases as the flame size also increases.

5.2 Radiant Heat Balance

An important parameter in the energy balance of a fire is the ratio ϕ_r of the radiant heat received by the fuel to the total radiant heat emitted by the flame.

The calculation of ϕ_r is very complicated because the flame emissivity depends on the flame thickness, and therefore, it results from numerical calculations of very complicated integrals.

For conical and cylindrical flames approximated values of ϕ_r are given by the expressions:

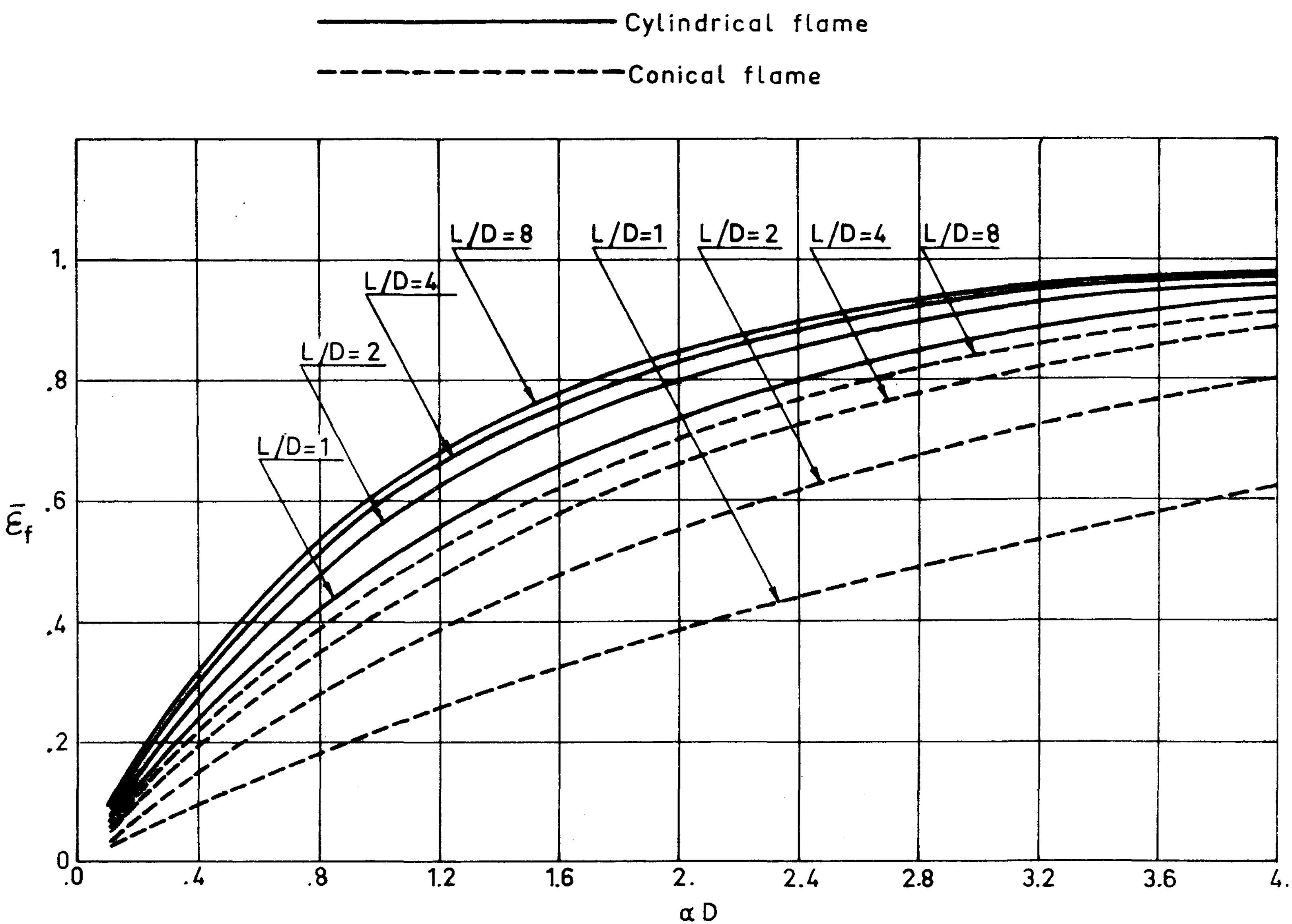


Fig.7. Theoretical Average Emissivity

$$\underset{\text{(conical)}}{\phi_r} = \frac{\int_0^{D/2} \left(1 - e^{-\alpha z}\right) 2\pi r dr}{2 \int_0^{D/2} \left(1 - e^{-\alpha z}\right) 2\pi r dr + \int_0^{L} \left(1 - e^{-2y\alpha}\right) 2\pi y\, dz} =$$

$$= \frac{1}{2 + 2 \left(\frac{L}{D}\right)^3 \dfrac{\frac{(\alpha D)^2}{2} + \alpha D e^{-\alpha D} + e^{-\alpha D} - 1}{1 - e^{-\alpha L} + \frac{(L\alpha)^2}{2} - L\alpha}} \tag{38}$$

$$\text{(cylindrical)} \quad \phi_r = \frac{1}{2 + 4 \frac{L}{D} \left(\dfrac{1 - e^{-\alpha D}}{1 - e^{-\alpha L}}\right)} \tag{39}$$

When the flame is very large it behaves as an opaque body. For this case we have:

$$\phi_r = \frac{\pi D^2/4}{A_{ft}} \tag{40}$$

$$\underset{\text{(conical)}}{\phi_r} = \frac{1}{1 + 2 \frac{L}{D} \sqrt{1 + \frac{D^2}{4\, L^2}}} \tag{41}$$

$$\text{(cylindrical)} \quad \phi_r = \frac{1}{2 + 4 \frac{L}{D}} \tag{42}$$

Fig.8 shows ϕ_r for conical and cylindrical flames of several sizes and shapes.

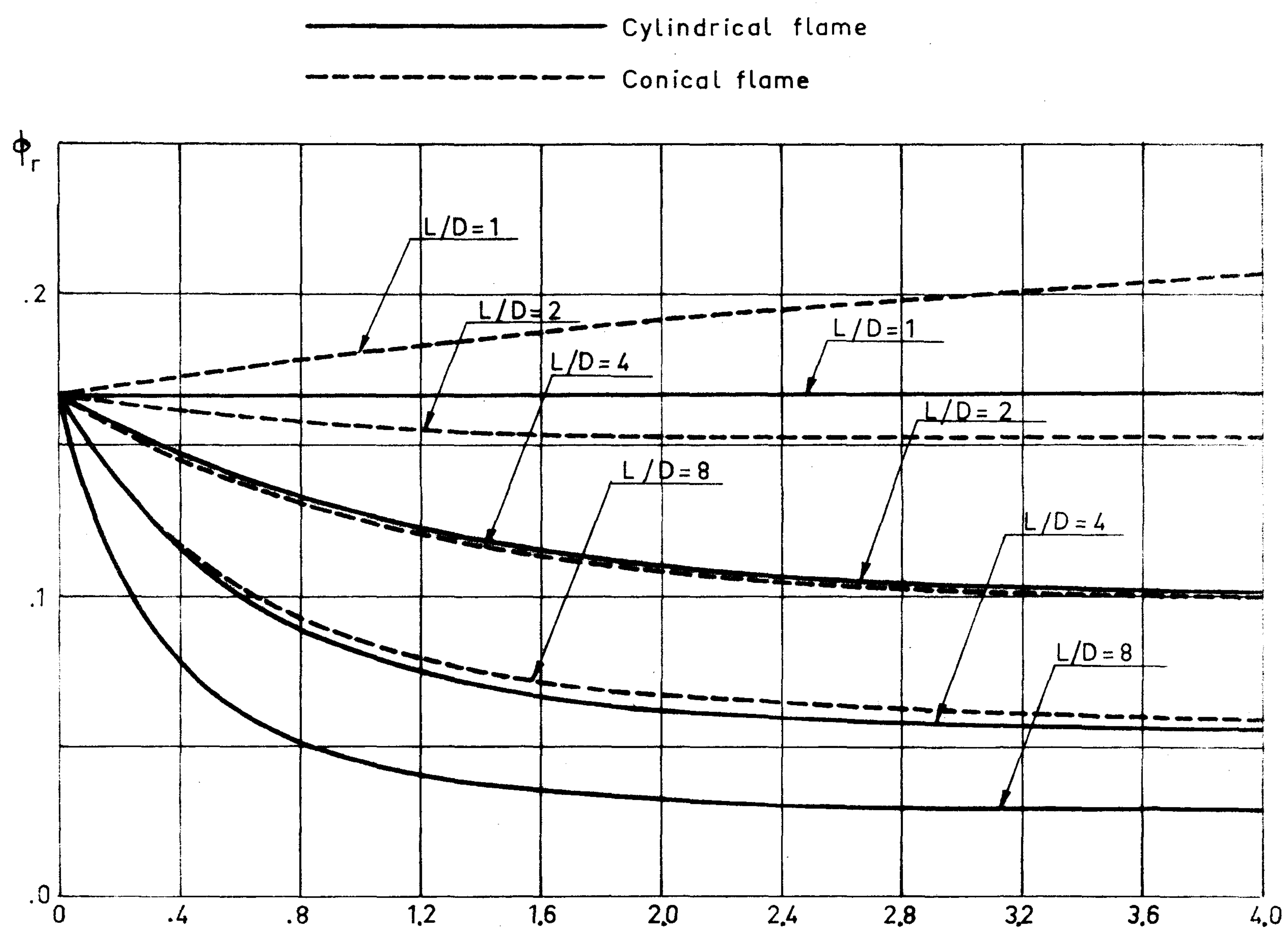


Fig. 8. ϕ_r- Ratio of Radiant Heat Received by the Fuel to the Total Radiant Heat Emitted by the Flame

6. ENERGY BALANCE

6.1 Stationary Process

Assuming that the surface reflectivity is equal to zero and that the radiant heat is absorbed by the fuel in a zero thickness layer, the fundamental equation results:

$$\bar{\dot{q}}_s = \bar{\dot{q}}_{cv} + \bar{\dot{q}}_{rf} = \bar{\dot{m}}_b \, q_1 + \bar{\dot{q}}_{cl} \qquad (43)$$

which expresses that the total heat reaching the fuel surface is equal to the heat consumed in evaporating the fuel plus the heat transmitted through conduction into the fuel.

Taking into (43) the values of $\bar{\dot{q}}_{cv}$, $\bar{\dot{q}}_{cl}$ and $\bar{\dot{q}}_r$, we obtain for the stationary process:

$$\dot{m}_b = \frac{1}{q_1}\left[\left(1-e^{-\alpha L_h}\right)\sigma T_s^4 \theta_f^4 + \frac{T_s \lambda_v}{z_{v,min}}(\theta_f - 1)\phi_v - \frac{T_s \lambda_1 (1-\theta_o)}{x_s}\,\frac{\nu e^{\nu}}{e^{\nu}-1}\right] \qquad (44)$$

which gives the burning rate as a function of flame size (L_h), flame temperature (θ_f), liquid fluid flow ν and physical fuel properties.

A numerical calculation of the different terms of Eq.(43) written in the form:

$$1 + \frac{\bar{\dot{q}}_{cl}}{\dot{m}_b q_1} = \frac{\bar{\dot{q}}_{rf}}{\dot{m}_b q_1} + \frac{\bar{\dot{q}}_{cv}}{\dot{m}_b q_1}$$

has been carried out for the case of the n-heptane with the following numerical data:

$x_s = 5$ cm $\quad T_f = 1000°$ C , $\quad T_o = 20°$C , $\quad \dot{m} = \dot{m}_b$

The radiation term has been obtained by substraction of the other terms. These terms are represented in Fig.9 as functions of $\dot{m}_b$. It may be seen that even for a very small value of the distance from the flame to the fuel surface ($z_v = 0{,}1$ cm) the heat transmitted through conduction tends rapidly towards zero as the burning rate augments.

The values of z_v required for combustion of a flow $\dot{m}_b$, for the case in which all heat is transferred through conduction, are shown in the following table

$\dot{m}_b$ gr/cm^2.sec	$0{,}5 \times 10^{-3}$	1×10^{-3}	$2{,}5 \times 10^{-3}$	10×10^{-3}
z_v cm	0,3	0,15	0,06	0,015

6.2 <u>Transient Process</u>

In the transient process the Eq. (43) is valid and gives

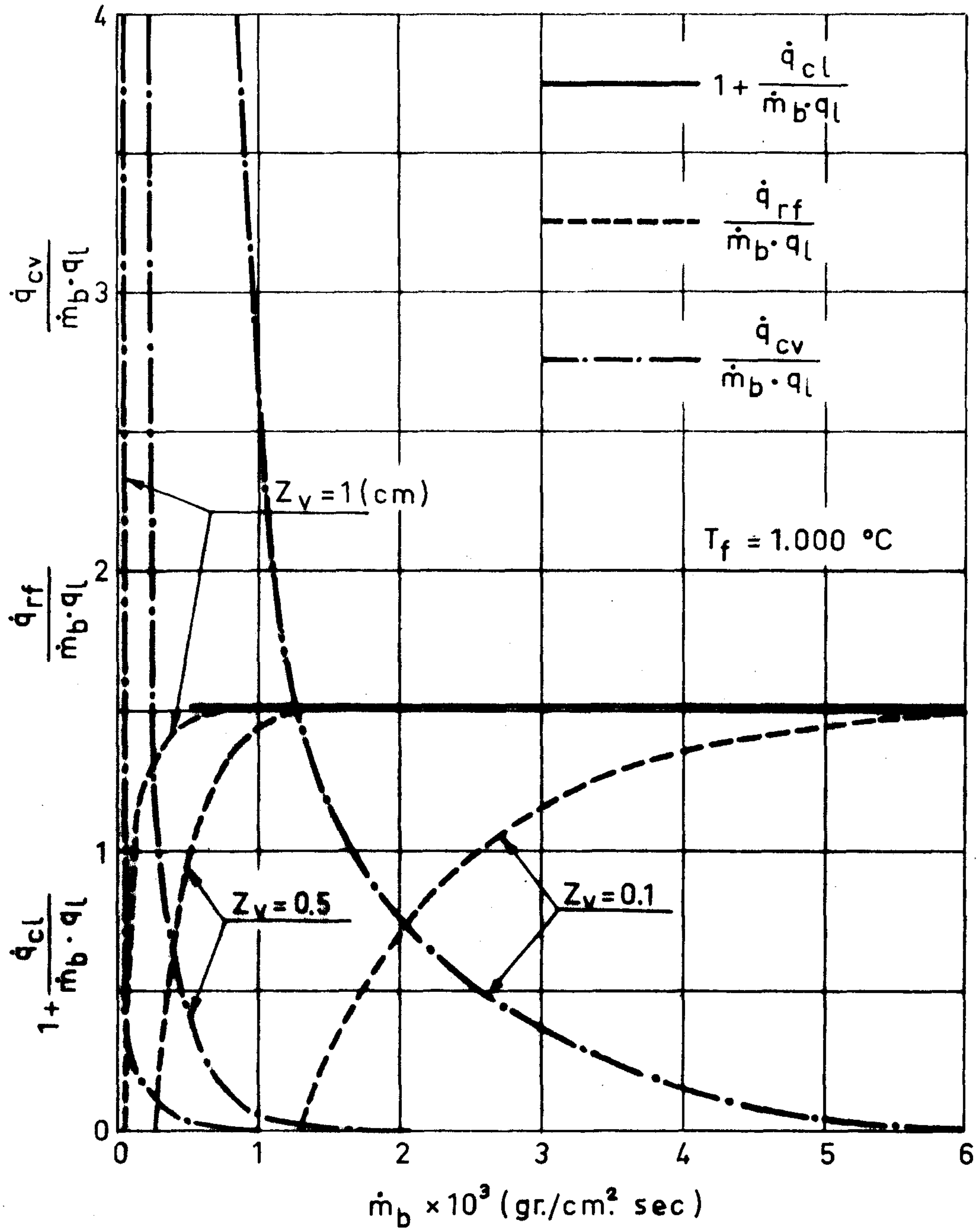


Fig.9. Theoretical comparison between radiant and convective heat received by the fuel

$$\dot{m}_b\ (t) = \frac{1}{q_1}\left[\left(1-e^{-\alpha L_h(t)}\right)\ \sigma T_s^4\theta_f^4 + \frac{T_s\lambda_v}{z_{v,min}}\ (\theta_f - 1)\phi_v(t)\ - \right.$$

$$\left. - \frac{T_s\lambda_1(1-\theta_o)}{x_s}\ \left(\frac{\nu e^\nu}{e^\nu - 1} + \sum_n \frac{8\pi^2 n^2}{\nu^2+4\pi^2 n^2}\ e^{-\frac{(4\pi^2n^2+\nu^2)\tau}{4}} \right)\right] \qquad (45)$$

θ_f and $z_{v,min}$ are practically time independent. In the transient process $L_h(t)$ increases, $\phi_v(t)$ decreases and the transient heat conducted through the liquid decreases.

7. THEORETICAL STUDIES IN SMALL VESSELS

In small vessels the flame is small and, therefore, the radiant heat flux received by the fuel is also small. The observed experimental increasement of the burning rate observed experimentally in this region of small diameters can not be explained correctly by using solely the preceeding conclusions. It may be due to the influence of the heat received by the fuel free surface from the flame through convection and especially because of the heat transmitted through the vessel lateral walls. These two processes have been independently studied and they may be superimposed as a first approximation.

7.1 Heat Convection through the Free-Surface of the Fuel

Heat transfer from flame to the fuel surface takes place through convection, the heat flux being in opposite direction to the fuel vapors flow.

For small vessels the flame is laminar and the volume occupied by the fuel vapors has an approximately conical shape.

It has been observed that, for laminar conditions, the height of the cone is of the same order of magnitude than that of the vessel radius.

Therefore, as a first approximation, both dimensions may be taken to be equal ($z_{v,max} = D/2$).

Eq.(27) gives the radial distribution of the heat received by the fuel surface through convection, which in turn, depends on the distribution of burning rate $\dot{m}_b$.

An approximation of the process, valid for small sized vessels, consists in disregarding radiation heat transfer to the fuel and in neglecting convective heat transfer within the fuel in a direction parallel to the fuel free surface.

These assumptions give the following equation for the balance of energy:

$$\dot{q}_{cv} = \dot{q}_{cl} + \dot{m}_b' \, q_l \qquad (46)$$

From this equation and taking into account Eqs.(27) and (15) when $\nu \gg 1$, which is the normal case, it is obtained:

$$\frac{\dot{m}_b' c_v \, (T_f - T_s)}{e^{\nu_v} - 1} = \dot{m}_b' . \left[q_l + c \, (T_s - T_o) \right] \qquad (47)$$

from which, it results:

$$\dot{m}_b' = \frac{\lambda_v}{c_v\ z_v} \ln \left[1 + \frac{c_v\ (T_f - T_s)}{q_1 + c\ (T_s - T_o)} \right] \qquad (48)$$

which gives the burning rate distribution as function of z_v, or with Eq.(28) as function of radius.

Burning rates ratios, taking into account (48), are:

$$\frac{\dot{m}_b'}{\dot{m}_{bo}'} = \frac{1}{1 + \left(\frac{z_{v,max}}{z_{v,min}} - 1\right)\left(1 - \frac{2r}{D}\right)} \qquad (49)$$

This expression is shown in Fig.10, in which the influence of vessel diameter may be observed.

Radiant heat flux and possible heat transfer within the liquid could make less sharp the burning rate curve.

The average value of burning rate is given by:

$$\dot{m}_b = \frac{4}{\pi D^2} \int_o^{D/2} \dot{m}_b' 2\pi dr$$

and considering Eqs. (48) and (28), it results:

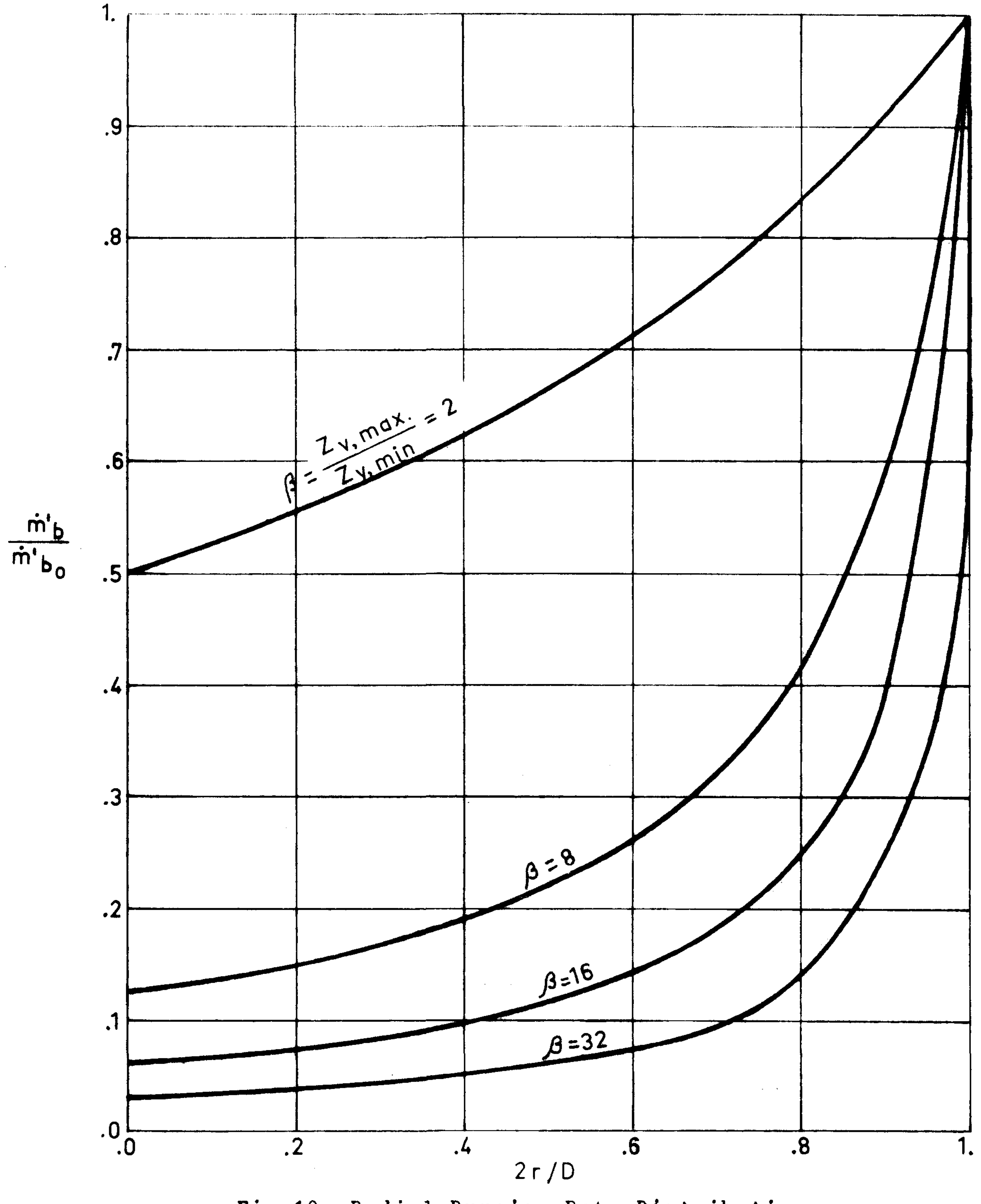


Fig.10. Radial Burning Rate Distribution

$$\dot{m}_b = \frac{\lambda_v}{c_v z_{v,min}} \ln \left[1 + \frac{c_v (T_f - T_s)}{q_1 + c (T_s - T_o)} \right] \frac{2}{(\beta - 1)^2} (\beta \ln\beta - \beta + 1) \quad (50)$$

Fig.11 shows the value of $\dot{m}_b / \dot{m}'_{b_o}$ assuming $z_{v,max} = D/2$. It may be seen that mean burning rate decreases when the vessel diameter augments.

7.2 Convection Through the Lateral Walls of Vessel

Heat exchange between the liquid fuel and vessel wall can be of certain importance, depending on the degree of cooling of the vessel. Through that wall, the fuel can receive heat from the flame or by the contrary, it may give off heat to the cooling fluid.

The expression of the heat transferred from the wall is:

$$\dot{Q}_w = \pi D x_s h \, (T_w - T_m) \quad (51)$$

in which T_w and T_m are the average temperatures of wall and fuel, and h is the convective heat transfer coefficient. For this coefficient the following expression may be taken:

$$N_u = \frac{hD}{\lambda} = A \left(\frac{\dot{m}_b D}{\mu}\right)^a \left(\frac{\mu c_p}{\lambda}\right)^b \left(\frac{D}{x_s}\right)^c \quad (52)$$

No complete information exists on the experimental values of coefficients A , a , b , and c for a wide range of variation of D/x_s . When $D/x_s >> 1$ the influence of heat exchange

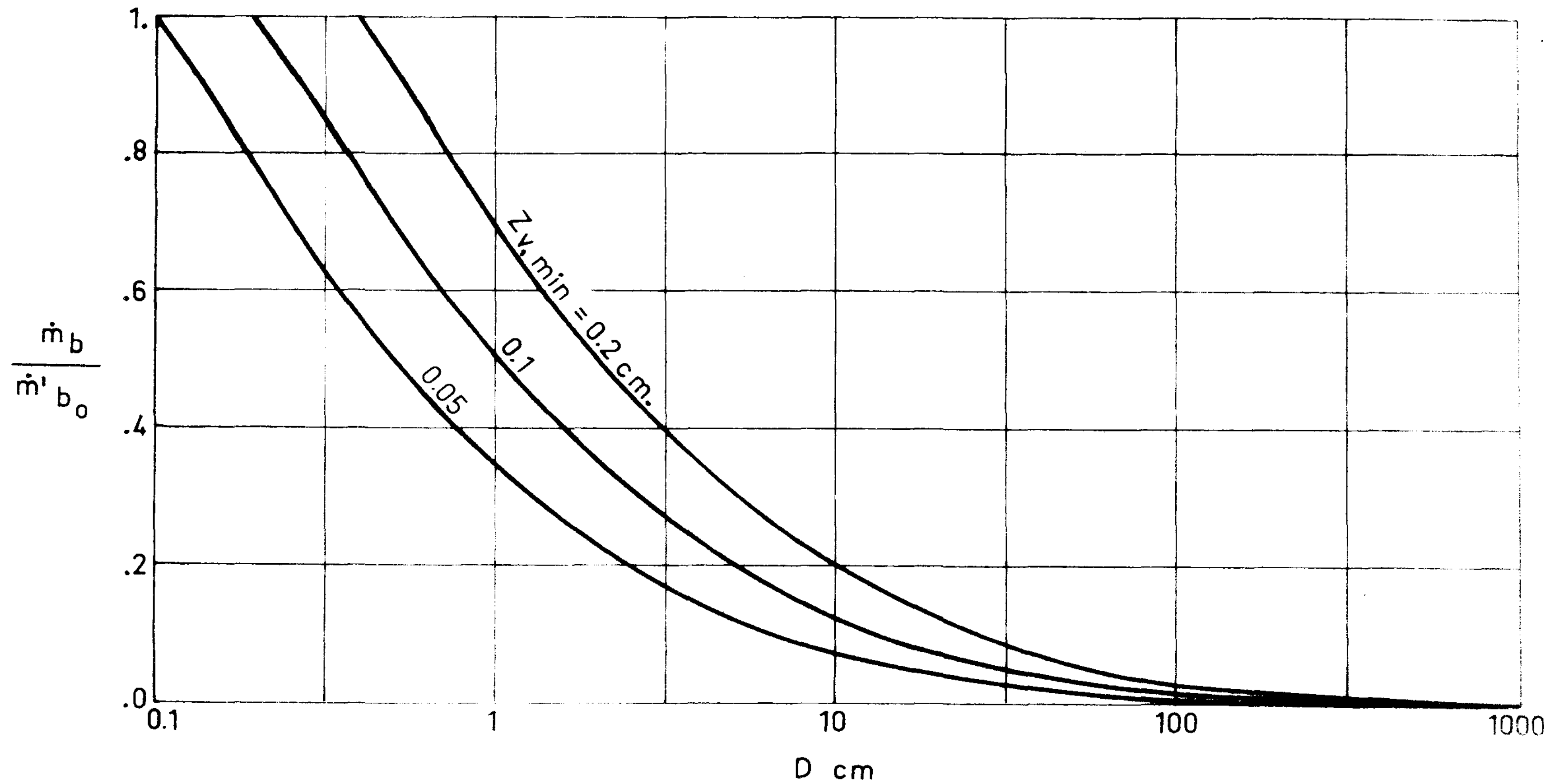


Fig.11. Small Vessels. Influence of **Convection through the Free** Surface of the **Fuel for** n-Heptane.

through the wall is small as compared with the other heat transfer mechanism and then it may be disregarded. On the other hand, when $D/x_s < 1$, the existing experimental data for tubes may be utilized as a first approximation, and this approximation improves as the ratio D/x_s becomes smaller, which is the case being studied.

Fuel motion in the vessel is laminar, and then, the following expression may be utilized:

$$N_u = \frac{hD}{\lambda} = 1,86 \left(\frac{\dot{m}_b D}{\mu}\right)^{1/3} \left(\frac{\mu c_p}{\lambda}\right)^{1/3} (D/x_s)^{1/3} \qquad (53)$$

in which fuel properties must be referred to the average temperature $T_m = (T_o + T_s)/2$.

Under these assumptions, the heat received by the fuel from the wall through convection, per unit area of free surface, and for a circular vessel, is given by

$$\dot{q}_w = \frac{4\,\dot{Q}_w}{\pi D^2} = 4\,.\,1,86\lambda\left(\frac{c_p}{\lambda}\right)^{1/3} \dot{m}_b^{1/3}\, D^{-4/3}\, x_s^{2/3}\,.\,(T_w - T_m) \qquad (54)$$

in which Eqs.(10) and (12) have been taken into account.

Considering this heat exchange, for small values of D , that is, when radiation is small, the equation expressing the energy balance is:

$$\dot{q}_w + \dot{q}_{cv} = \dot{m}_b \left[q_l + c\,(T_s - T_o)\right] \qquad (55)$$

The left hand side of this expression implicity contains burning rate $\dot{m}_b$, and its determination from that equation is difficult. A first approximation is to assume that no interaction exists between the direct convection flame-fuel and the indirect one through the wall, and then to superimpose both effects, that is to say:

$$\dot{m}_b = (\dot{m}_b)_{\dot{q}_w=o} + (\dot{m}_b)_{\dot{q}_{cv}=o} \qquad (56)$$

In this expression $(\dot{m}_b)_{\dot{q}_w=o}$ is given by Eq. (50) and $(\dot{m}_b)_{\dot{q}_{cv}=o}$ is derived from Eq. (54), being:

$$(\dot{m}_b)_{\dot{q}_{cv}=o} = \left[\frac{7{,}44\lambda(c_p/\lambda)^{1/3}(T_w - T_m)\ D^{-4/3}}{q_l + c\ (T_s - T_o)}\right]^{3/2} \qquad (57)$$

Fig.12 shows the results obtained from this expression for n-heptane.

Actually, the convective effect from the wall should be in some degree larger than that given by Eq. (57), since $\dot{m}_b > (\dot{m}_b)_{\dot{q}_{cv}=o}$, and the convective effect on the fuel surface should be somewhat smaller than that given by Eq. (50), since $\dot{m}_b > (\dot{m}_b)_{\dot{q}_w=o}$.

It may be pointed out that the convective effect from the wall can be positive or negative according to the sign of $T_w - T_m$.

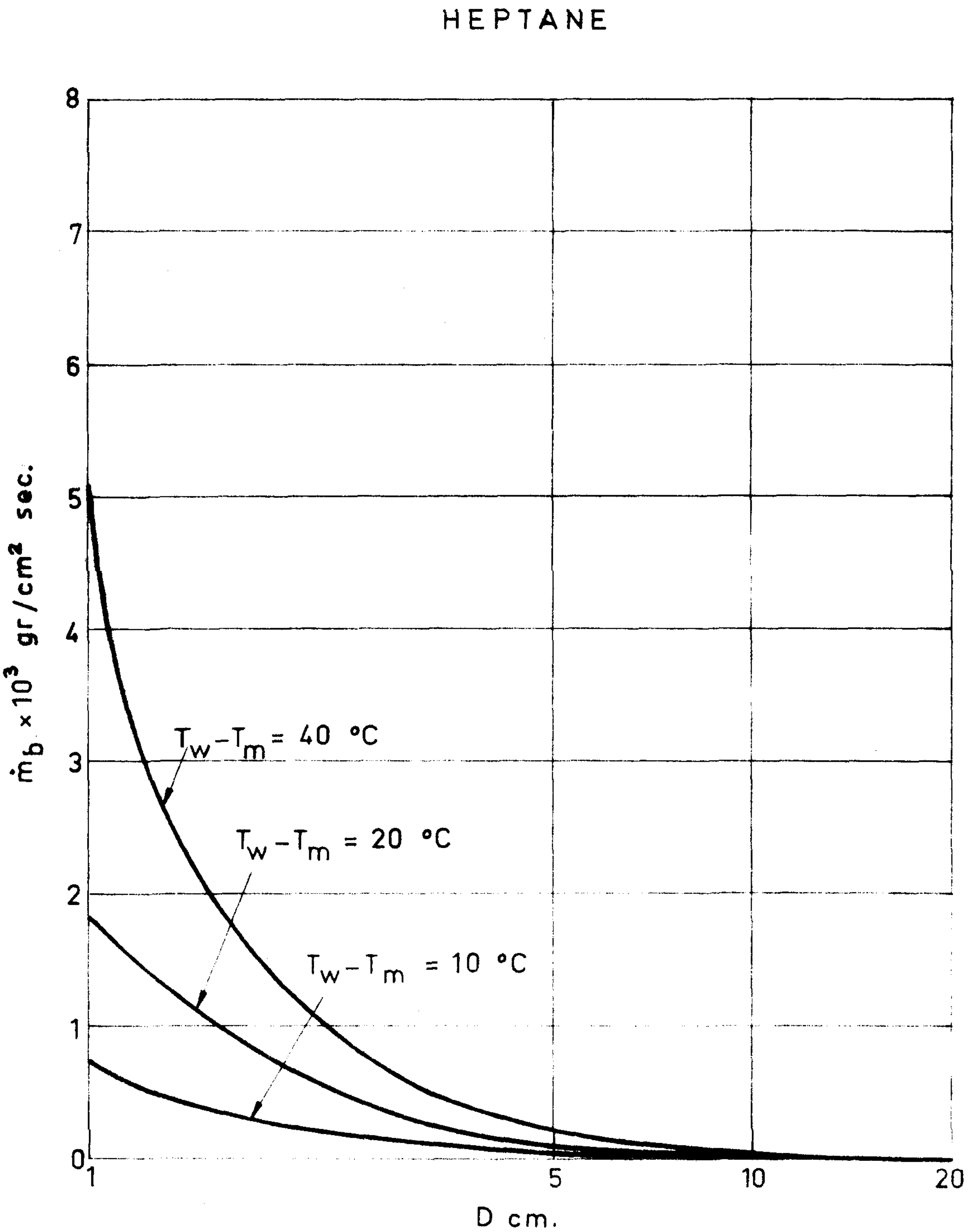


Fig.12. Small Vessels. Influence of Convection through the Lateral Wall for n-Heptane

II. EXPERIMENTAL STUDIES

1. RESEARCH FACILITIES

1.1 General Description

A research facility was prepared for studying open fires, which is represented in Fig.13.

The burner is a cylindrical open vessel in which the fuel is kept at a constant level by means of a small tank, where the fuel level is mantained with a small overflow tube placed in the center of the vessel.

Fuel is supplied from a tank and fuel consumption is measured by means of a constant pressure volumetric system.

Several types of vessels were tested until a model was finally selected. This vessel is represented in Fig. 14 and it has been constructed in several sizes.

Fuel depth was chosen as a compromise between opposite conditions. Large amounts of fuel in the vessel make difficult the attainment of stationary conditions, because burning rates keep increasing for a very long time. On the other hand, small depths of fuel may originate important temperature gradients within the fuel, parallel to the fuel surface.

Fuel temperature is measured by means of three thermocouples which can be set at different distances from the fuel surface. The heat lost by the liquid fuel transferred to

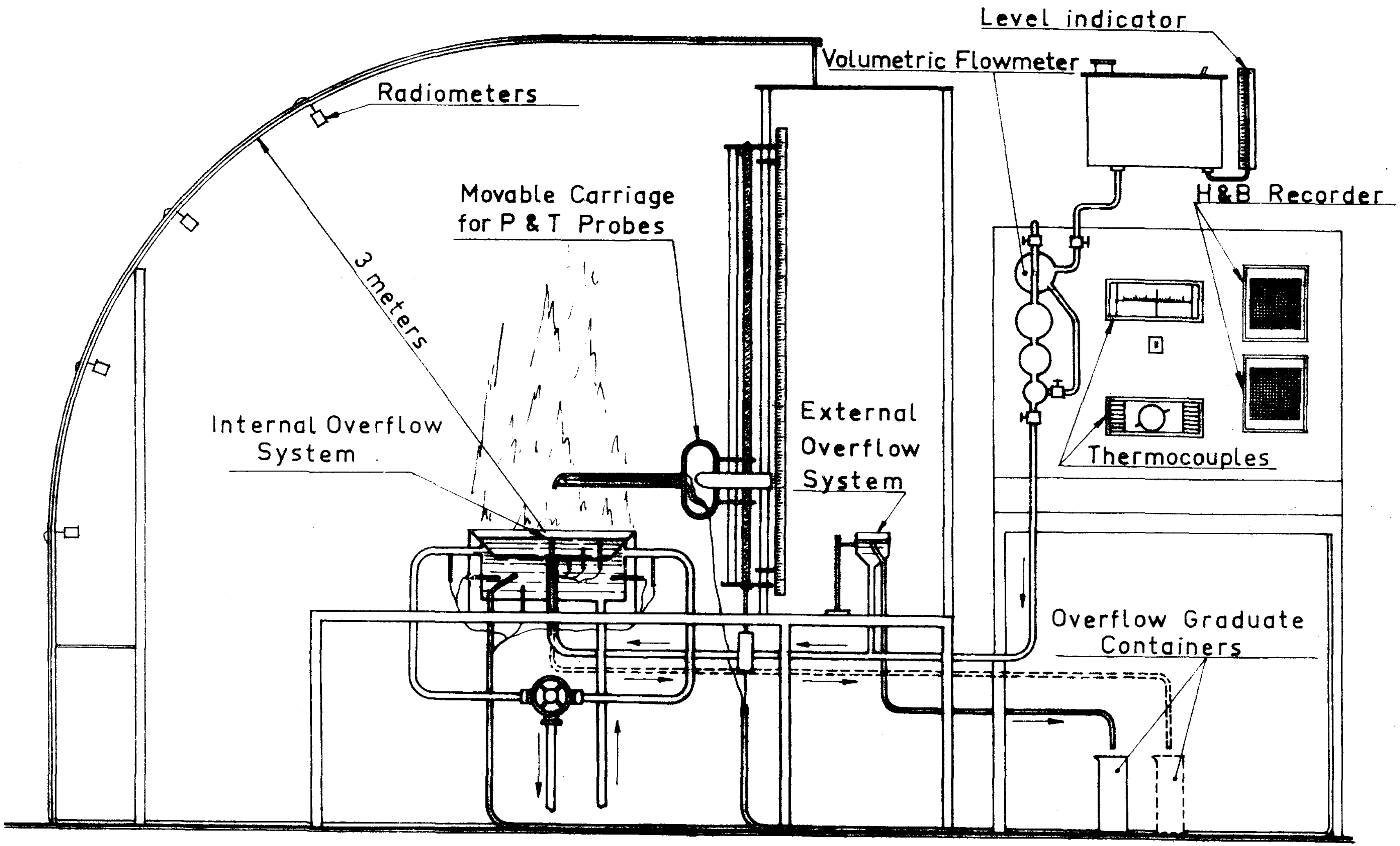


Fig.13. **Research Facility for Studying Open Fires**

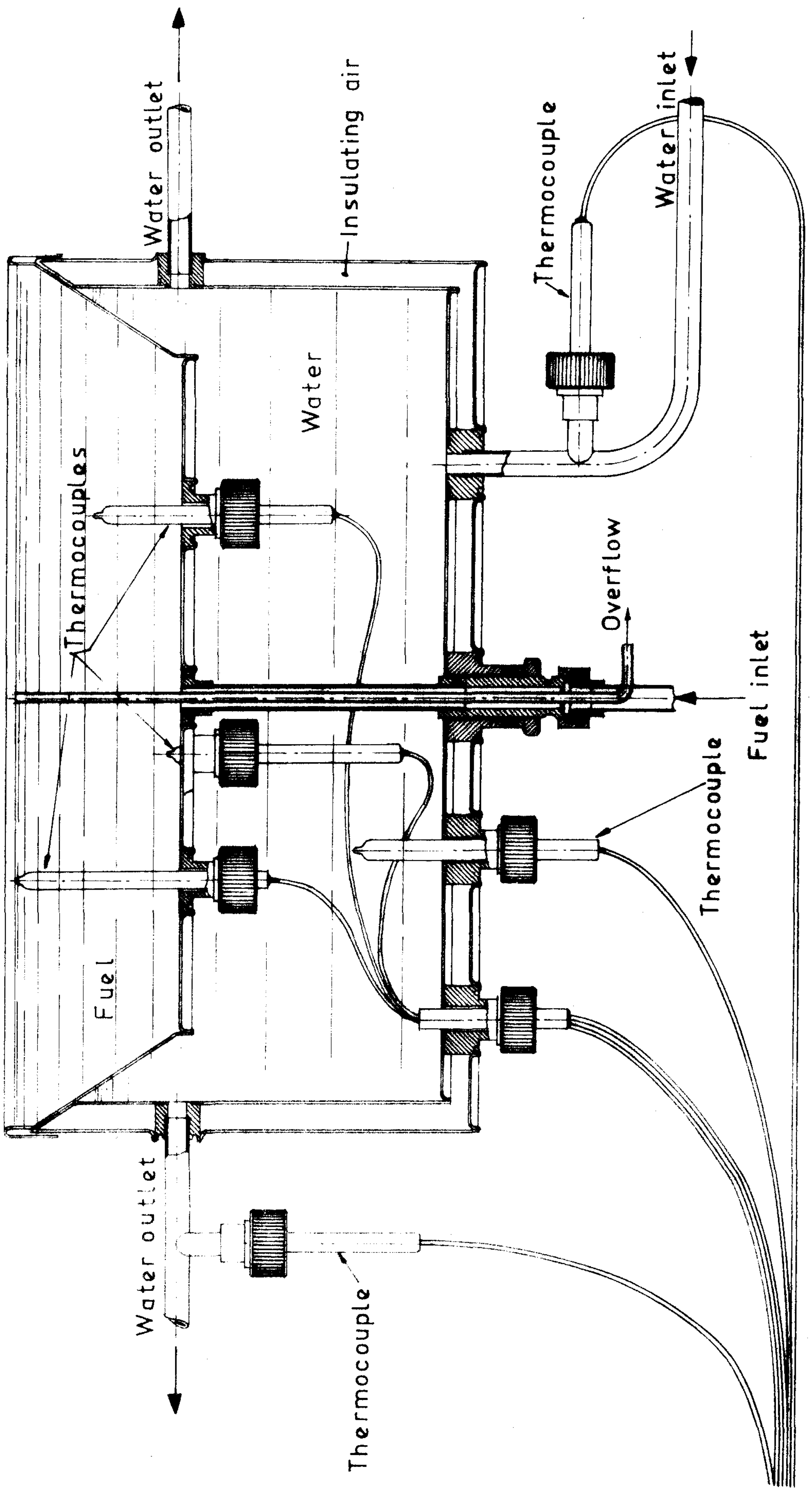


Fig.14. Diagramatic Sketch of Vessel.

cooling water is determined by measuring the increment of the temperature of this cooling water which surrounds the fuel container.

Vessels of 12.5, 25 and 50 cm in diameter have been constructed. All of them have very narrow rims in order to minimize heat transfer from the flame to the fuel via the vessel rim.

A grid with different sized holes has been placed at the bottom of the vessel, in order to have a more uniform distribution of the incoming fuel.

A ring with four Kipp and Zonen Radiometers has been placed over the fire, and their signals are registered in a six channels Hartman and Braun pen recorder.

A moving system of probes has been placed. Several pressure and temperature probes have been attached to a moving carriage, which can be displaced by remote control along horizontal and vertical axis. This system permits measurements of flame temperatures and gas velocities at continuously variable location during the experiments.

At first, the research facility was placed in an engine test stand, but there were some slight air drafts which produced undesirable motion of the flame, difficulting considerably the measurements.

Therefore, a tower of an engine test stand was specially conditioned for the study of open fires. By means of a brick wall and a steel screen the tower was isolated, and a roof with lateral holes was also constructed. A diagram of the tower is shown in Fig.15.

1.2 Small vessels

The research facilities was modified for small vessels as shown in Fig.16, utilizing two vessels of 250 mm and 72 mm in diameter divided in three annular compartments of equal area.

1.3 Fuels

Several types of fuels are used for studying open fires. The purpose of using different fuels is to study the influence on the process of parameters such as heat of combustion, heat of evaporation, flame luminosity (soot formation) etc. Gasoline, kerosene and n-heptane have been utilized.

It was very desirable to have a fuel in which its properties could change in a continuous form, specially its heat of combustion, because it would enable to have a continous variation of flame temperature. Therefore, a fuel was sought which could mix with water in all proportions and with similar boiling temperature, in order that the mixture would keep the same composition during its evaporation and combustion processes.

It was considered that the most suitable commercial product was the ethylene-dioxide (dioxane) $C_4H_8O_2$.

Boiling temperature of dioxane is 101°C and a density of 1.033 at 20°C.

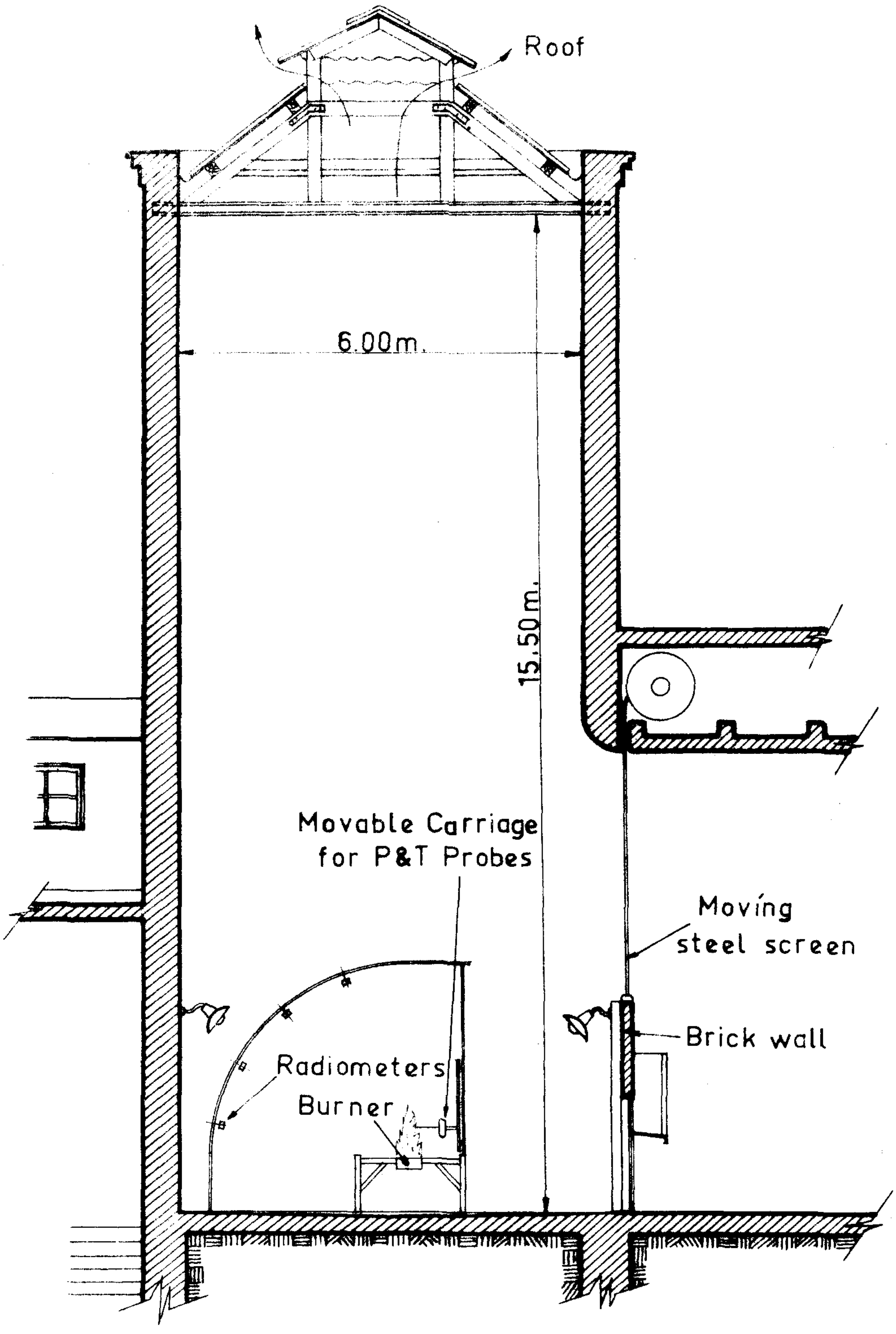
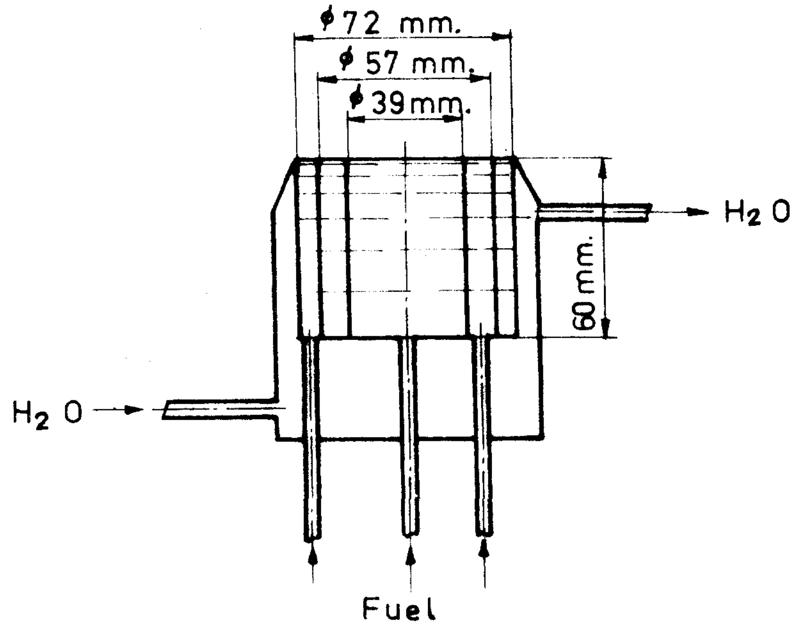


Fig.15. Test Stand for Studying Pool Fires.



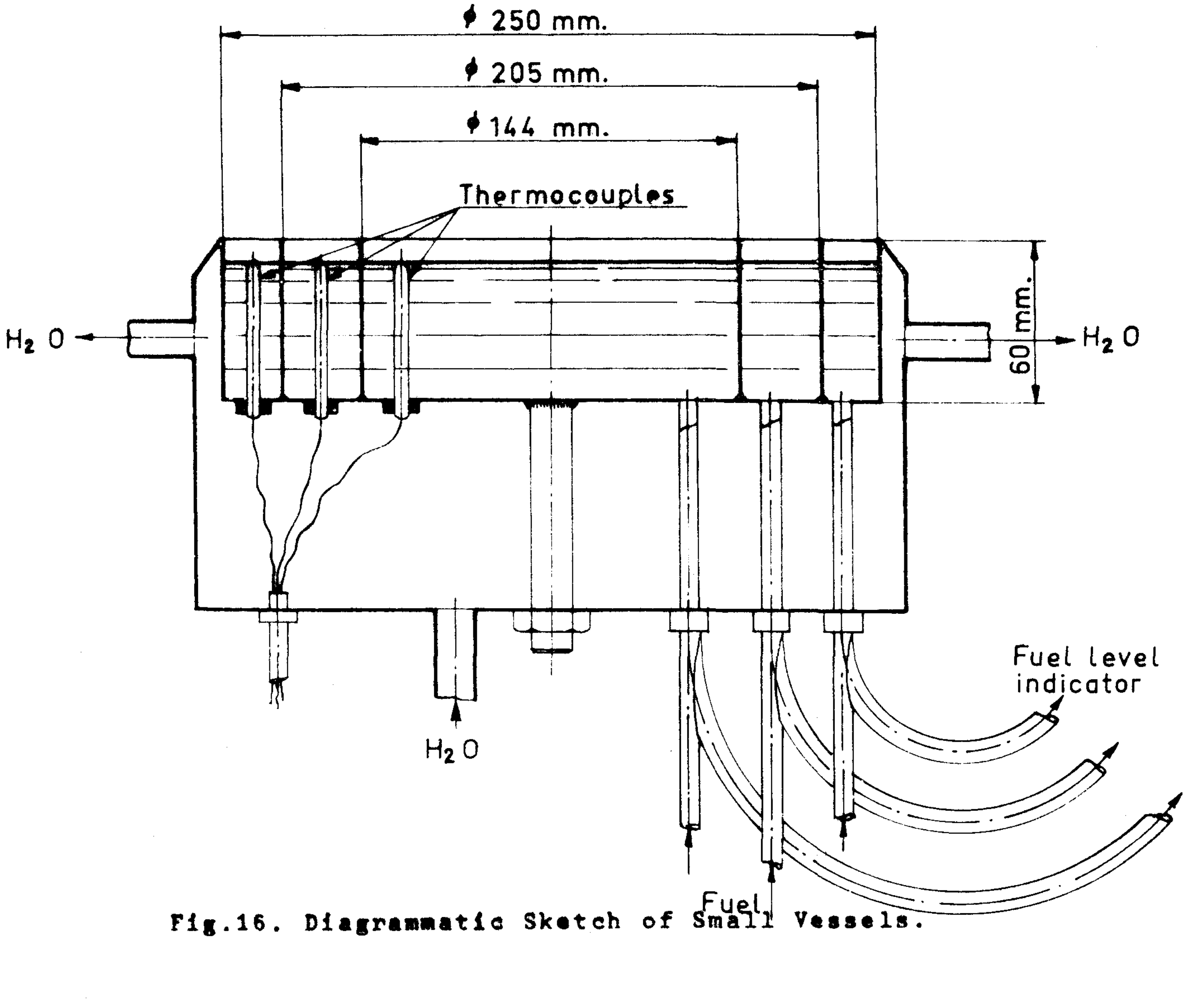


Fig.16. Diagrammatic Sketch of Small Vessels.

The fuel is expensive (about 90 pesetas a kilogram), but it offers special advantages for several types of studies.

In the first place, mixtures of 25%-50% water and 75%-50% dioxane were burned and analyzed at different times. It was found that the initial proportion dioxane-water was mantained practically constant during combustion.

The heat of combustion of the dioxane was not found in the literature. Therefore it was determined at the Chemistry Laboratory of the Institute giving the following values:

100/100 Dioxane,	q_r = 6,260 cal/gr.
Dioxane + 25% water in volume,	q_r = 4,720 cal/gr.
Dioxane + 50% water in volume,	q_r = 3,110 cal/gr.

Fig.17 shows the values of q_l , q_r , and q_r/q_l for several mixtures dioxane-water.

Photographs of flames produced by the three mixtures in a vessel of 30 cm in diameter are shown in Fig.18. It may be observed the difference in length and luminosity of the flames.

2. RESEARCH PROGRAM

An extense experimental program was conducted, comprising the following measurements:

1°) Burning rates and temperature profiles within the fuel.

2°) Radiant heat emitted by the flame.

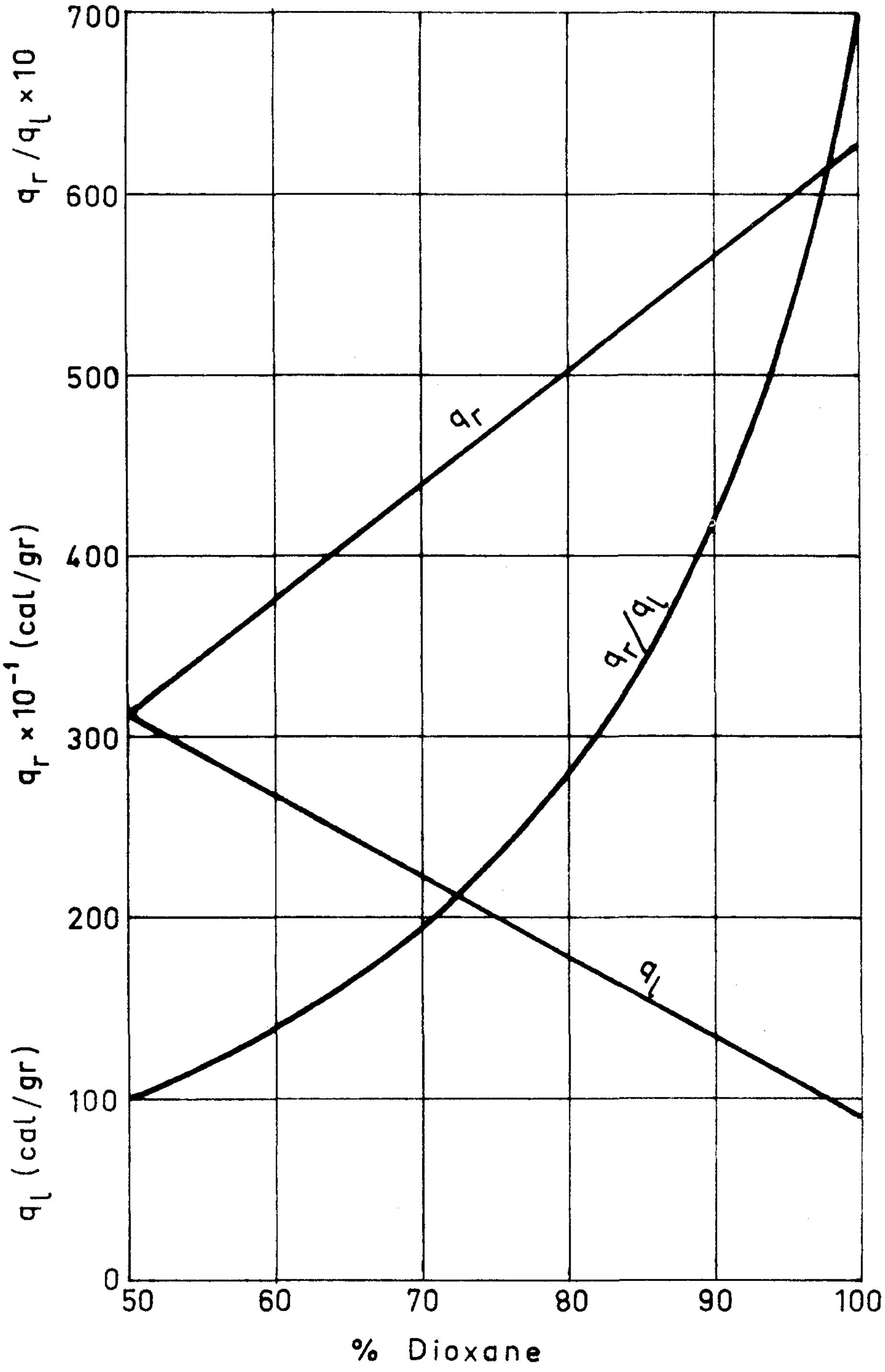


Fig.17. Physical Properties of Dioxane-Water Mixtures.

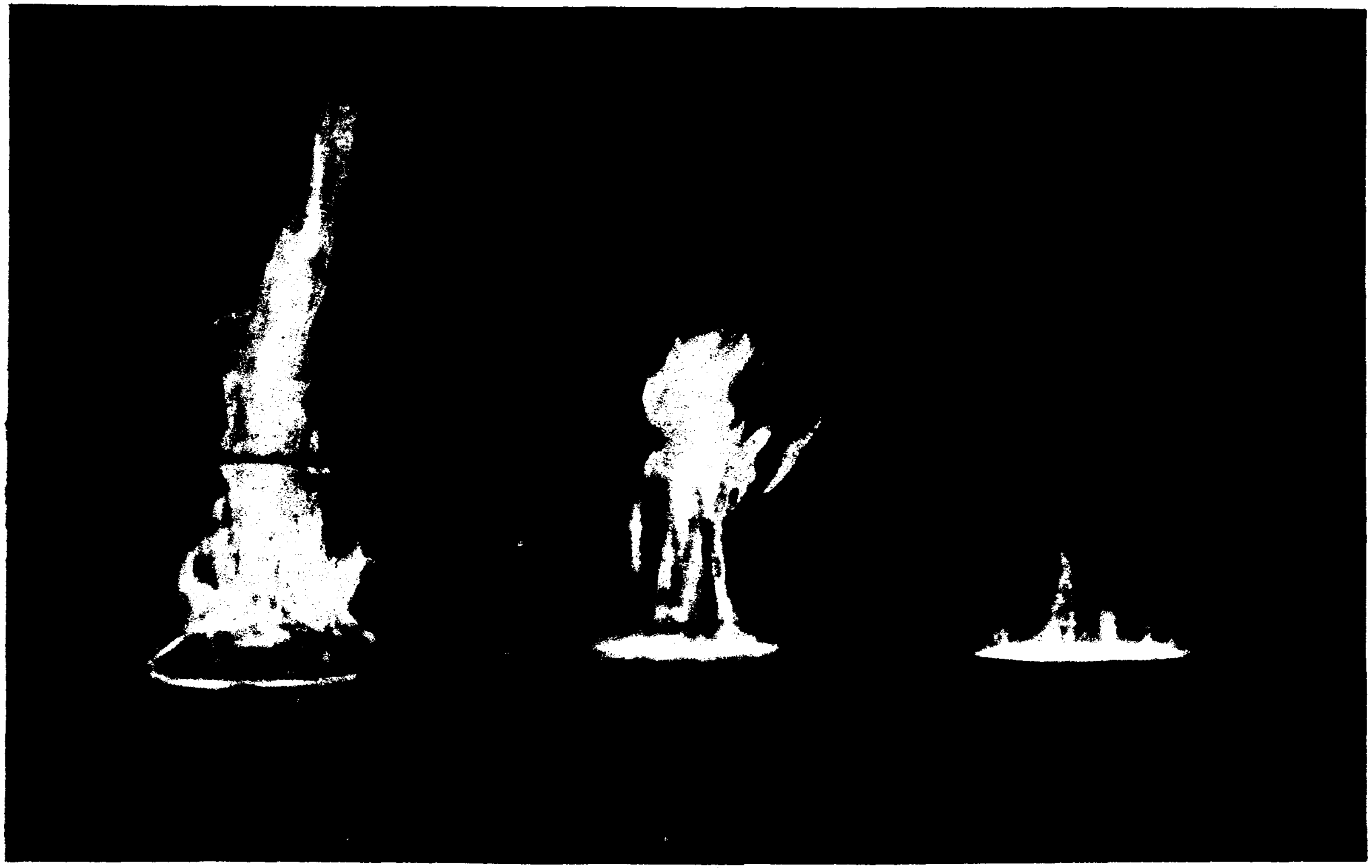


Fig.18. Photographs of Dioxane Flames.

3°) Flame size and flame lateral surface (from movie pictures)

4°) Flame temperatures at several locations.

The influence of vessel size, vessel depth and flame size on the problem has been studied. For a given vessel and burning the same fuel, flame size can be varied considerably by changing the overflow rate.

For small vessels the research program has been devoted to the study of the radial distribution of the burning rate and of the influence of the vessel cooling.

3. EXPERIMENTAL RESULTS

Figs. 19, 20 and 21 show three examples of the type of measurements which have been carried out.

The values of the area of lateral surface of the flame have been obtained by means of the expression $A_l = \pi A_{fr}$, where A_{fr} is the cross-section area of the flame. The value of the heat radiated by the flame per unit time to the surrounding atmosphere Q_{re} has been calculated by numerical integration from radiometers measurements.

Flame temperatures were measured by means of five moving Ni-Ni Cr thermocouples.

Fig.22 shows the measured temperatures, as well as the mean flame shape for n-heptane burning in a vessel of 25 cm in diameter. Each temperature is an average value of

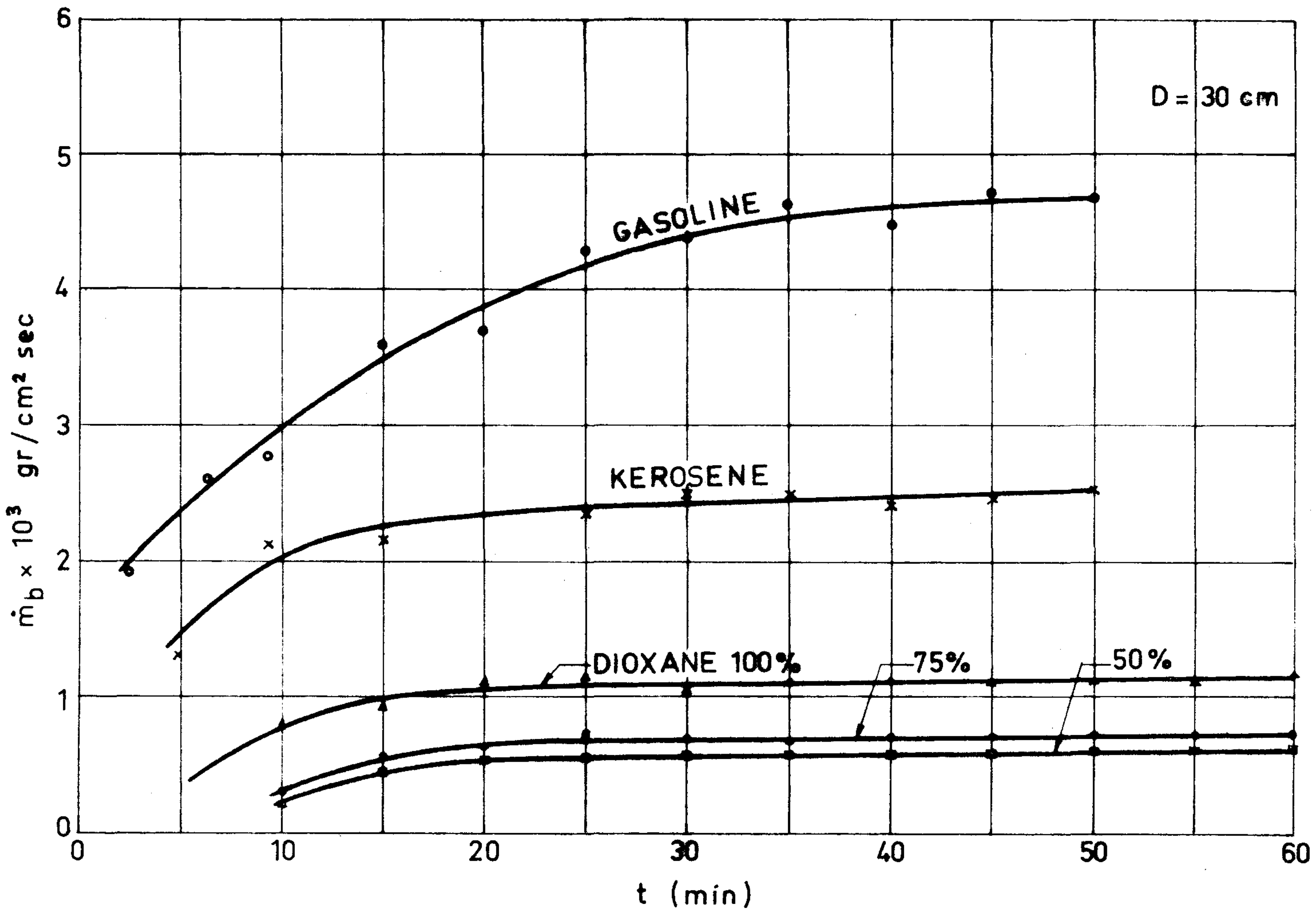


Fig.19. Burning Rates for Several Fuels

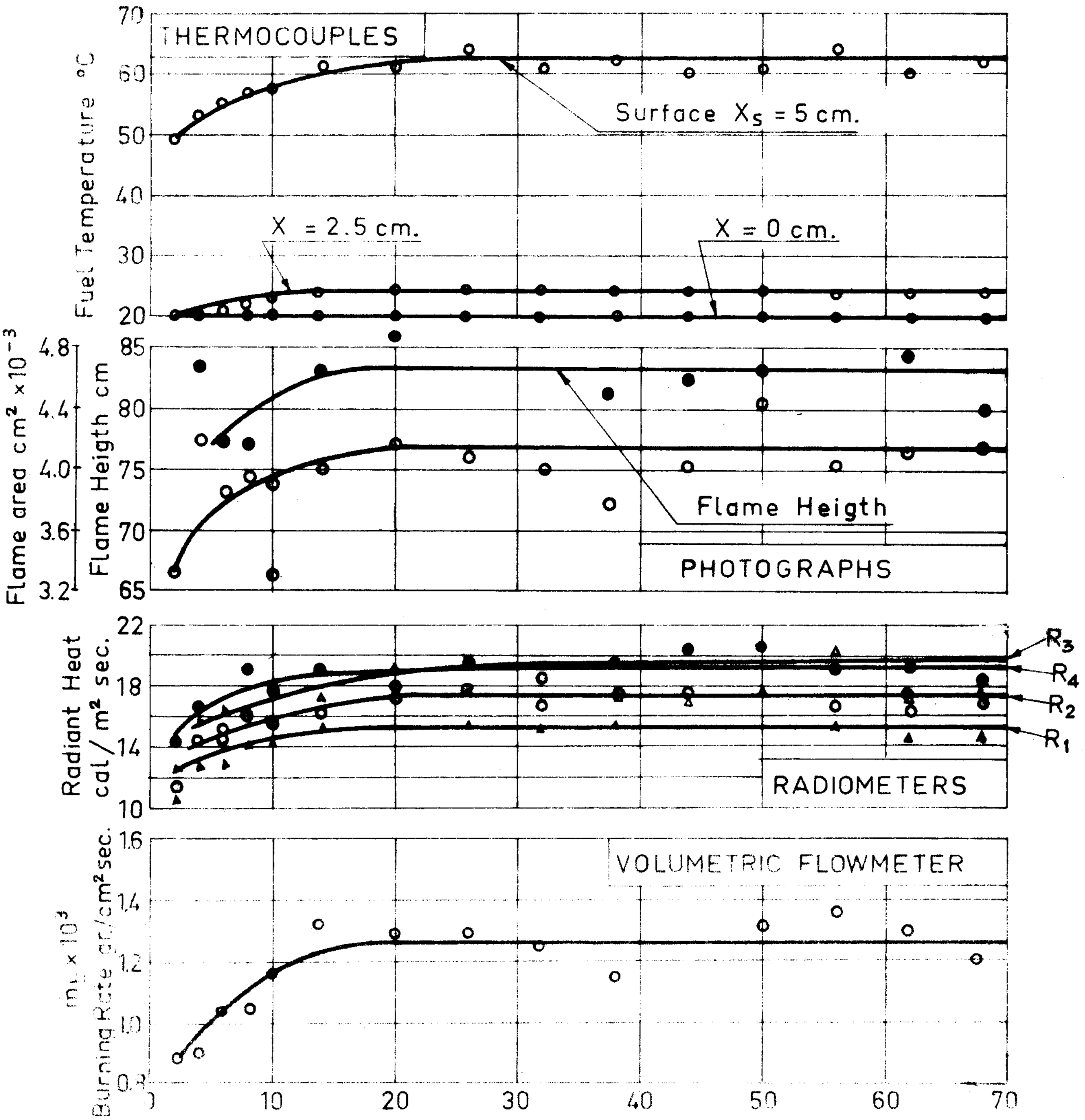


Fig.20. Experimental Results, n-Heptane, D = 25 cm.
$\dot{m}$ = 5,32 10^{-3} gr/cm^2.sec.

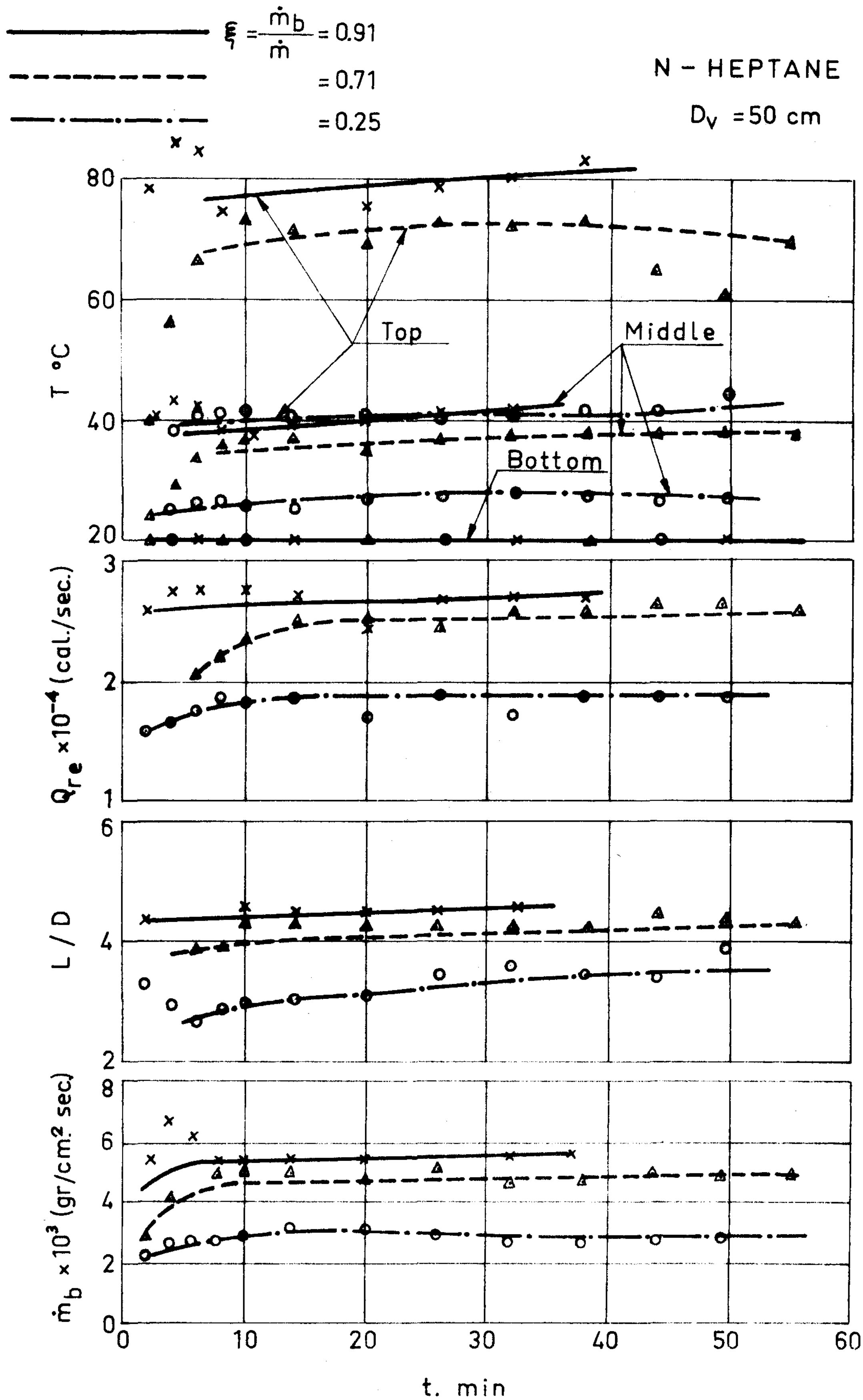


Fig.21. Experimental Results. n-Heptane, D = 50 cm.

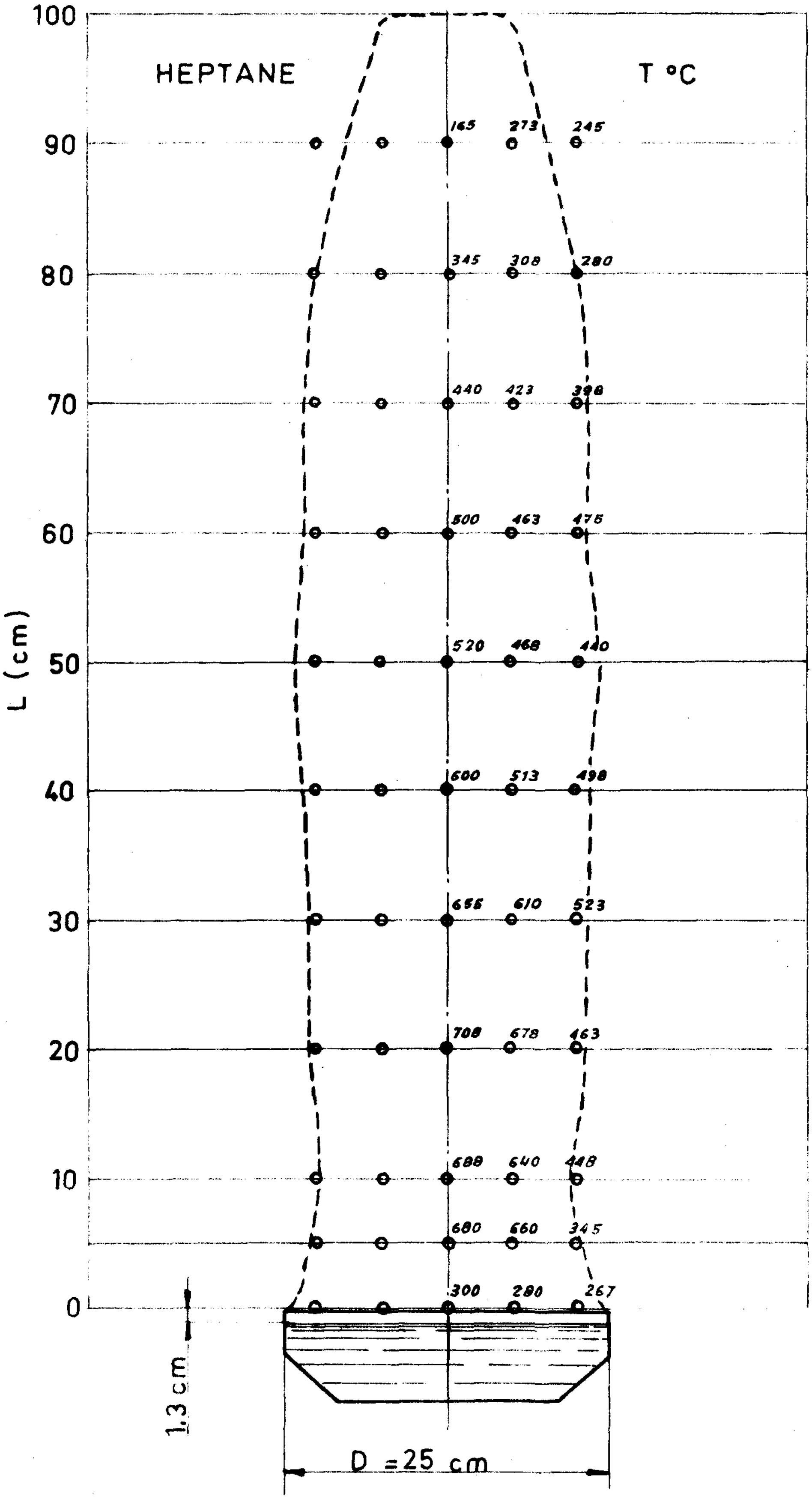


Fig.22. Flame Temperature Distribution, n-Heptane.

six measurements.

The measured temperatures near the flame borders and within its upper region are not actual values, because during part of the time the thermocouples did not remain within the flame due to the oscillatory motion of this flame. Temperature distribution in the central zone is similar to that found by Fons et al[3].

It can also be deduced from the figure the existence of a fuel vapors zone in the vicinity of the fuel surface, zone in which the temperature is low.

Fig.23 shows similar temperature measurements for dioxane, the values being of the same order of magnitude.

Flame temperatures measured by means of thermocouples are not very accurate because of soot formation, radiation errors and flame fluctuations. Optics measurements give average values (in time and location) with values somewhat higher than those measured by thermocouples[1] . However, accurate measurements of temperatures at a given point in a large turbulent flame is still a problem to be solved.

3.1 Influence of the Overflow

The experimental results obtained are shown in Figs. 24 and 25. Each point of these figures is the average value of about 20 measurements which were taken after combustion reaches stationary conditions.

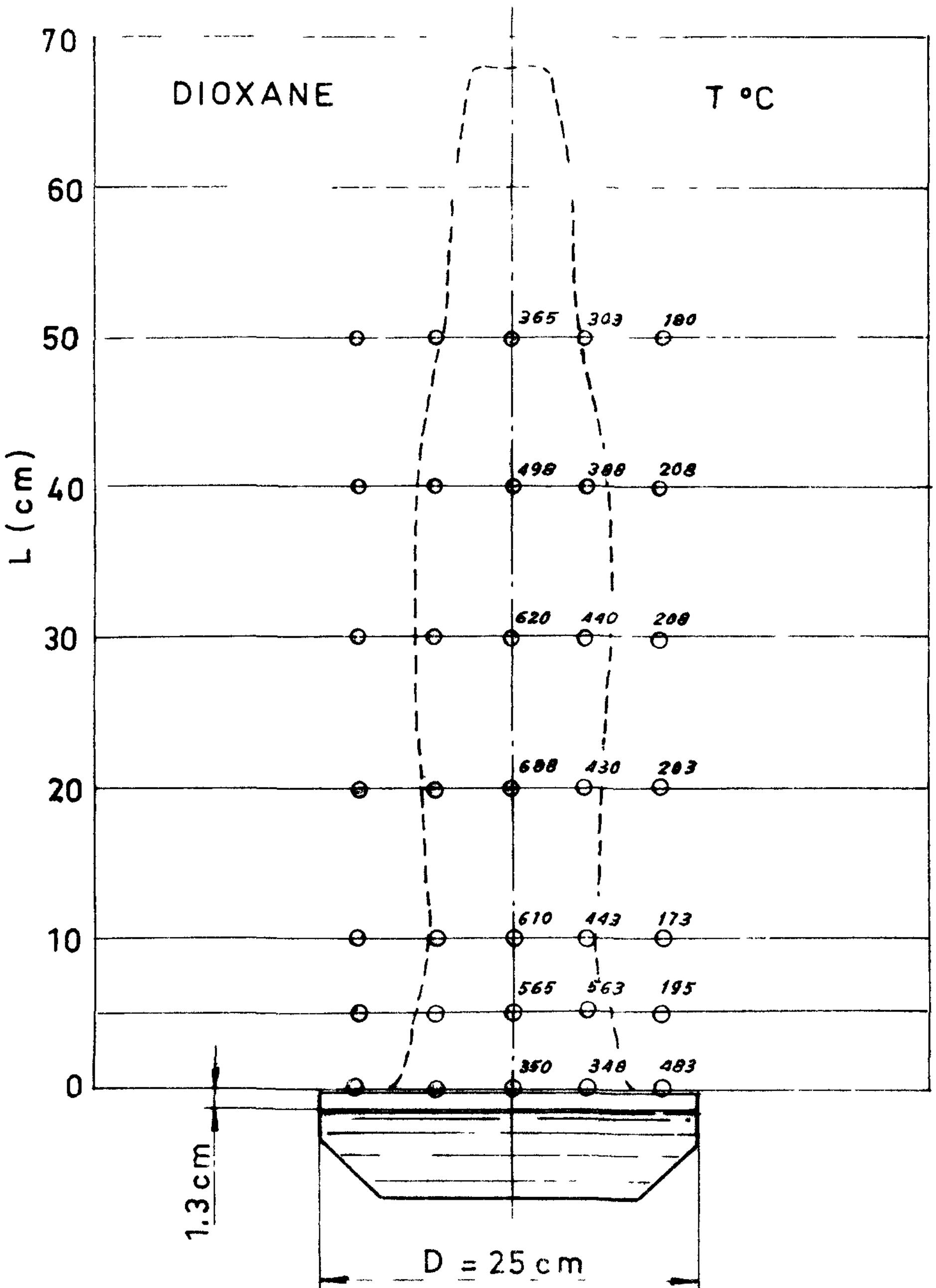


Fig.23. Flame Temperature Distribution. Dioxane.

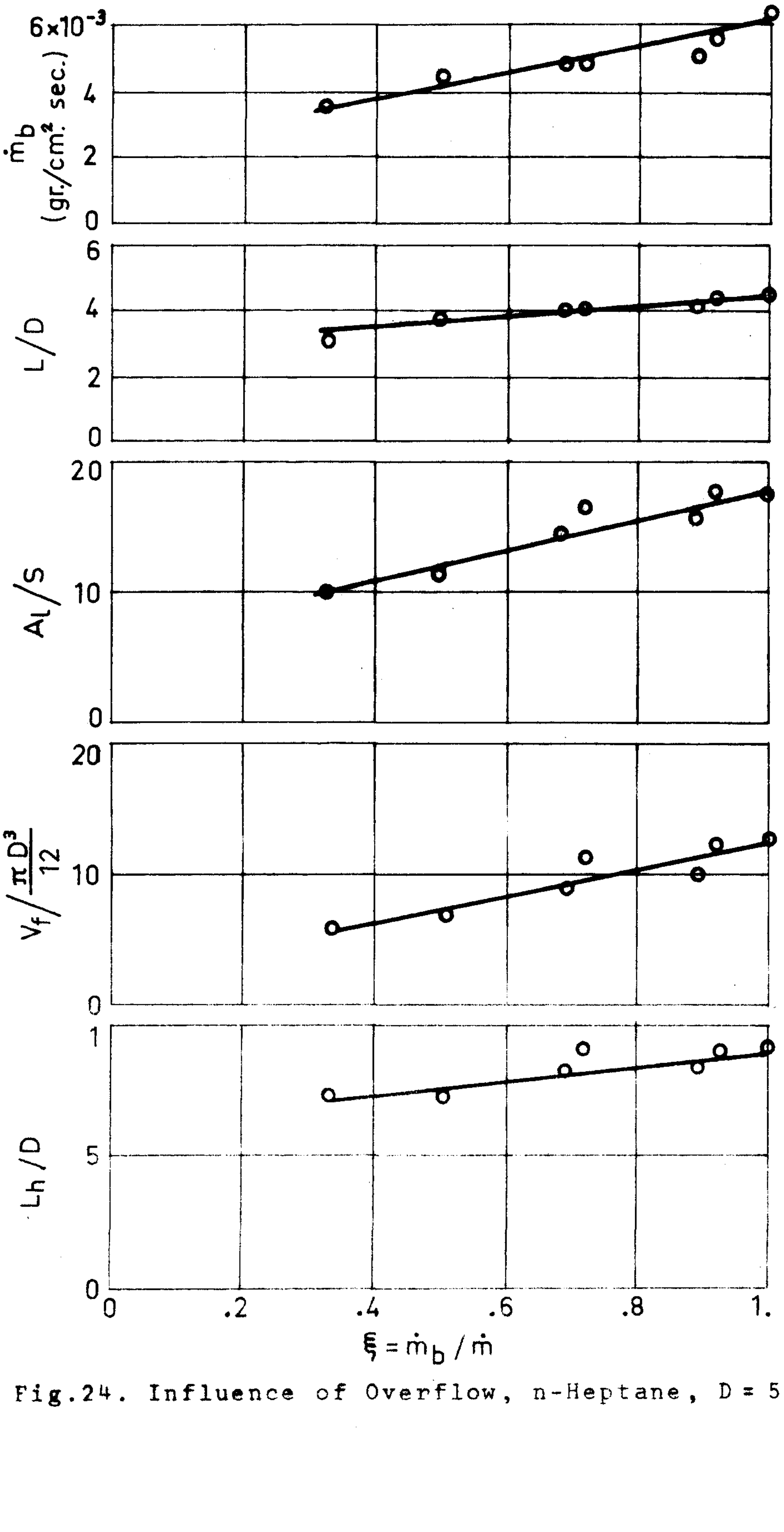


Fig.24. Influence of Overflow, n-Heptane, D = 50 cm.

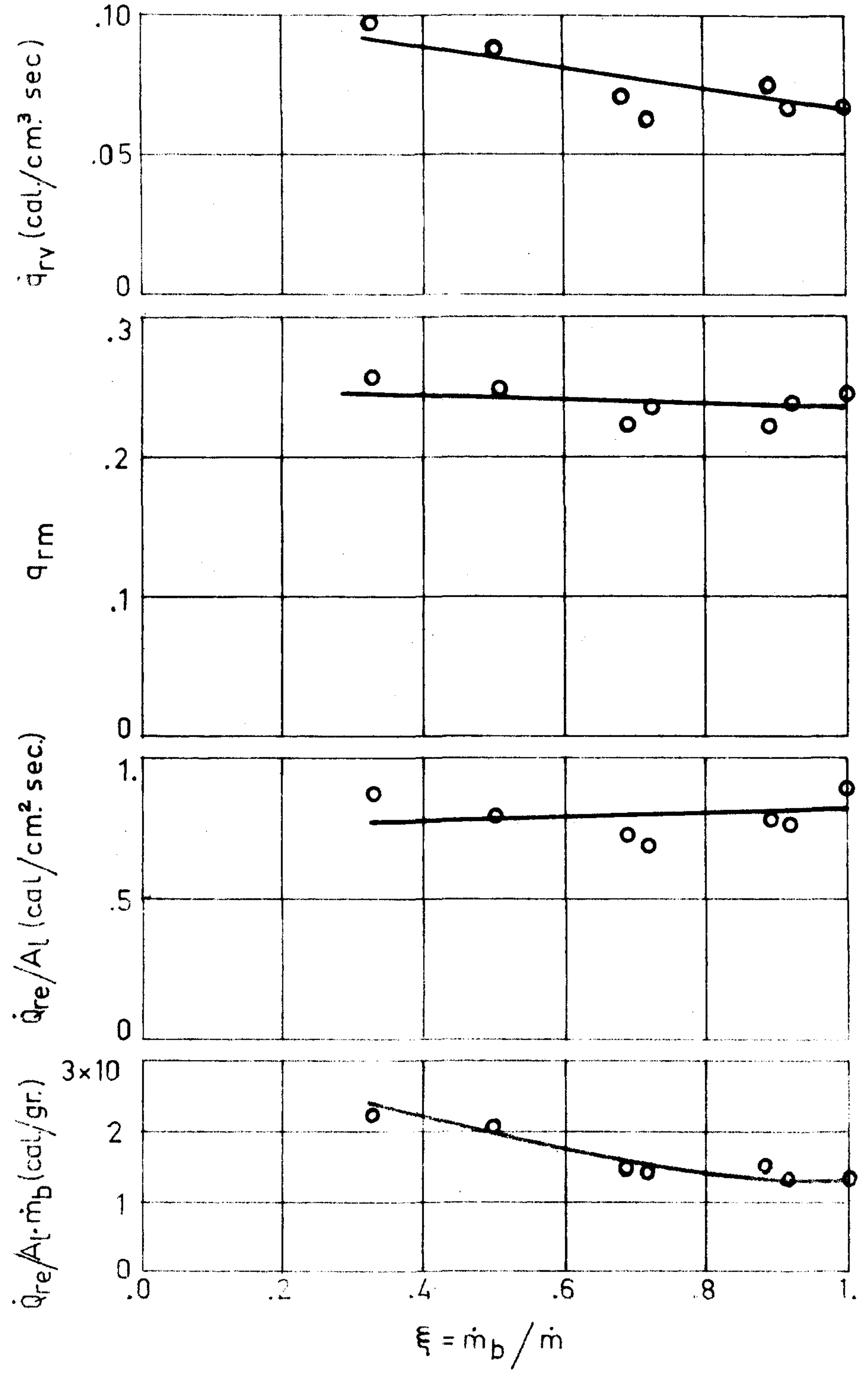


Fig.25. Influence of Overflow, n-Heptane, D = 50 cm.

3.2 Influence of the Vessel Diameter

The experimental results obtained with n-heptane are shown in Figs. 26 and 27. In Fig.26 it may be observed that burning rate increases as vessel's diameter augments, except in the region of small values which has comprised till the value corresponding to $D = 3$ cm.

Another conclusion, derived from that figure is that flame volume is proportional to the third power of vessel's diameter and that L_h is proportional to that diameter.

Total radiant heat $\dot{q}_{rv}$ per unit flame volume, (Fig. 27), is constant and the total radiant heat per unit mass of burnt fuel q_{rm} is also constant.

Figs. 28 and 29 show the experimental results obtained with dioxane, whose physical and chemical properties differ appreciably from those of n-heptane.

However, these results are similar to those obtained with n-heptane. It may be pointed out that $\dot{m}_b$ increases more slowly and that $\dot{q}_{rv}$ decreases as the diameter augments.

3.3 Influence of the Type of Fuel

Figs. 30 and 31 show the experimental results obtained with several dioxane-water mixtures. It may be observed that $\dot{m}_b$ increases as the proportion of water in the mixture decreases, as it might be expected. Dioxane flames do not extend over the whole fuel surface as shown in Fig.23. Therefore, for small flames all numerical results in connection with

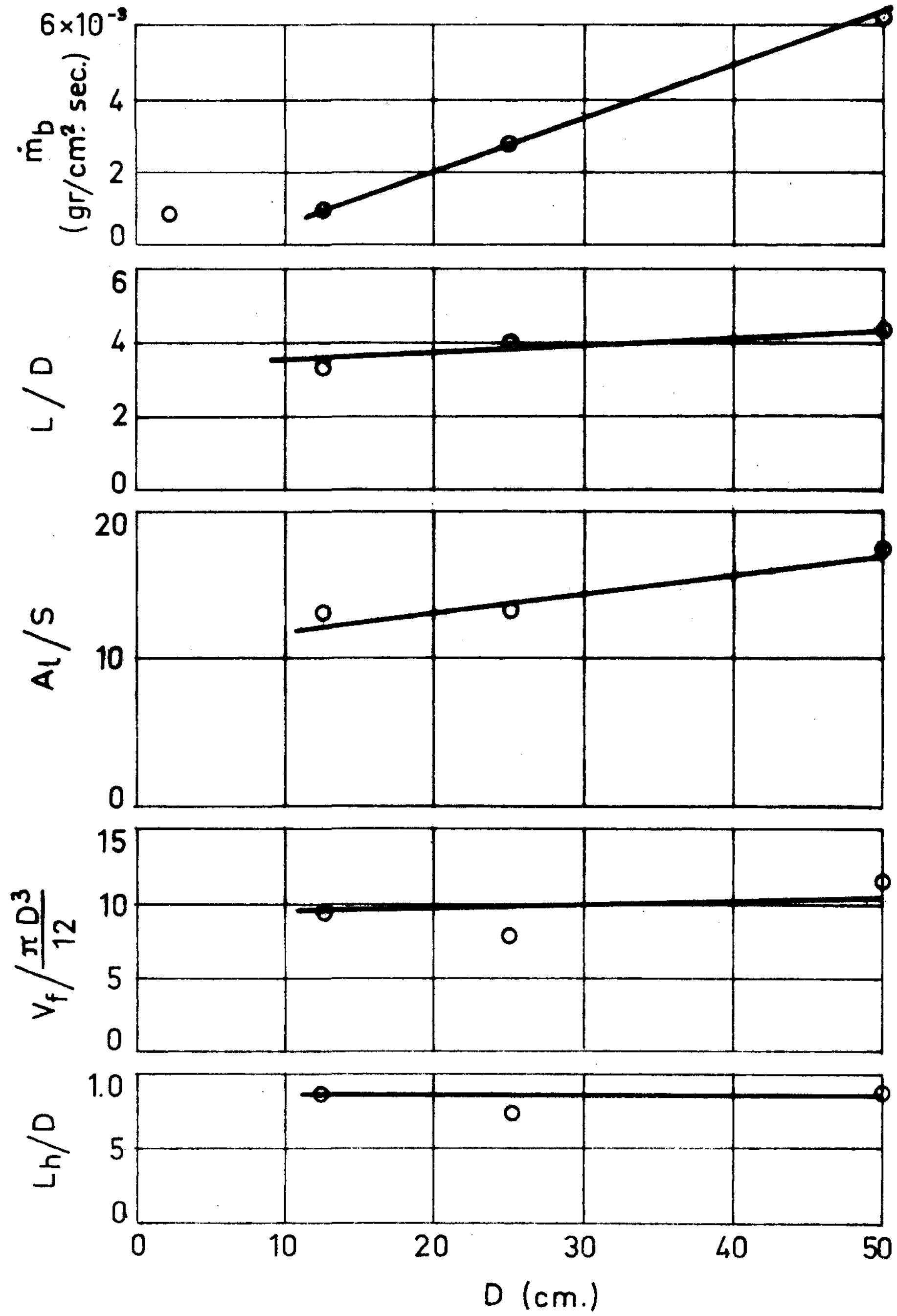


Fig.26. Influence of Diameter. n-Heptane, ξ = 1.

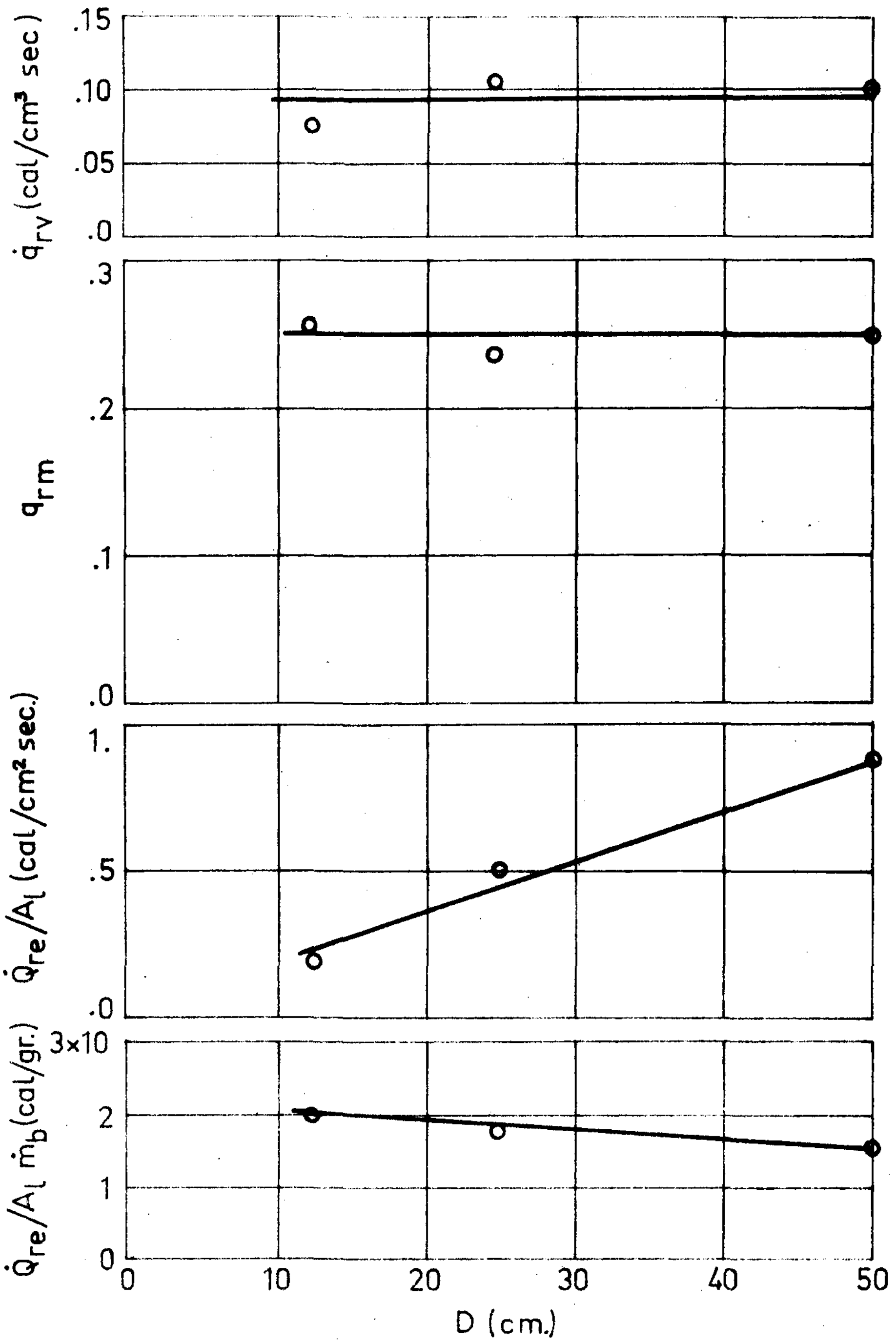


Fig.27. Influence of Diameter. n-Heptane, ξ = 1 .

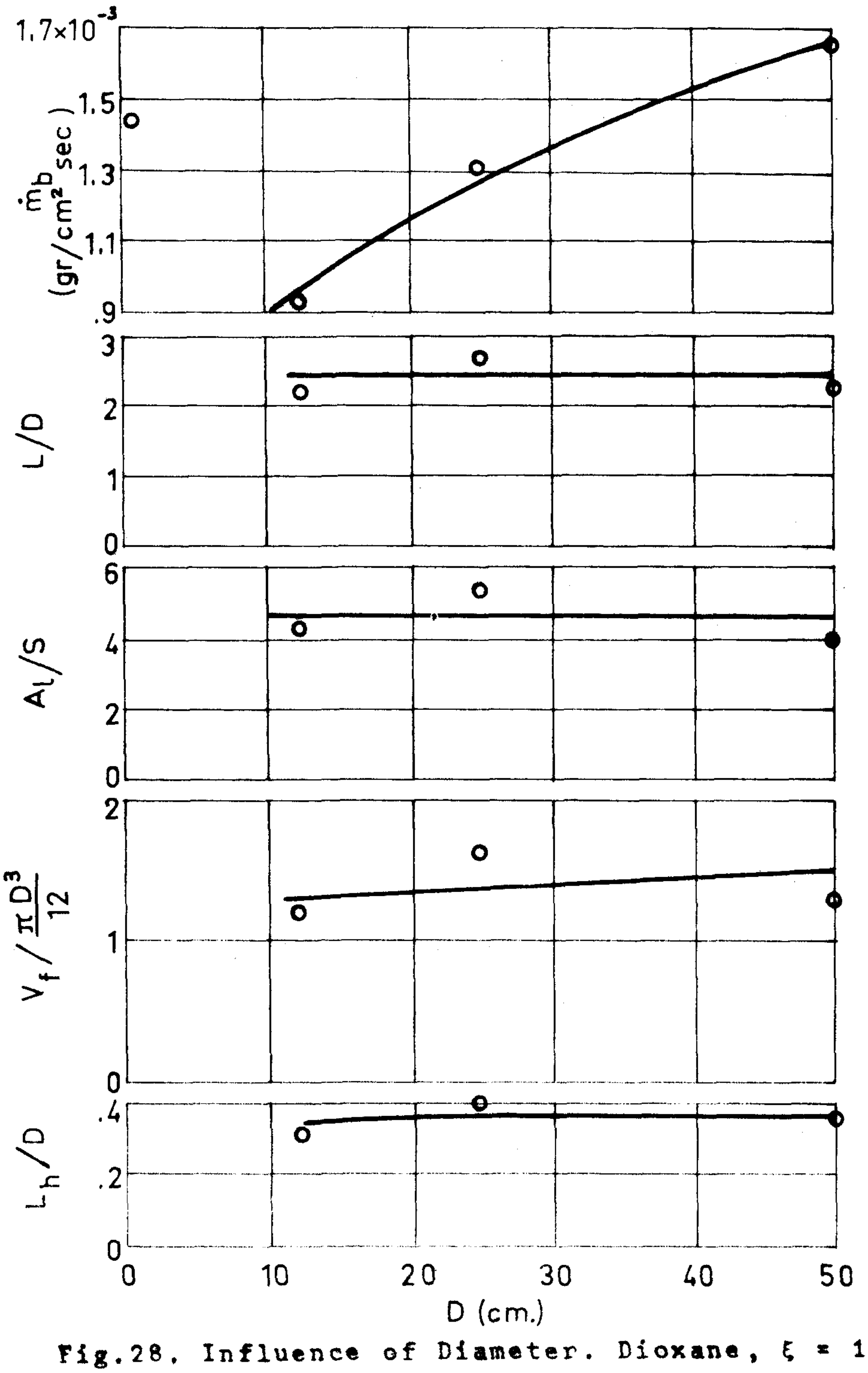


Fig.28. Influence of Diameter. Dioxane, ξ = 1 .

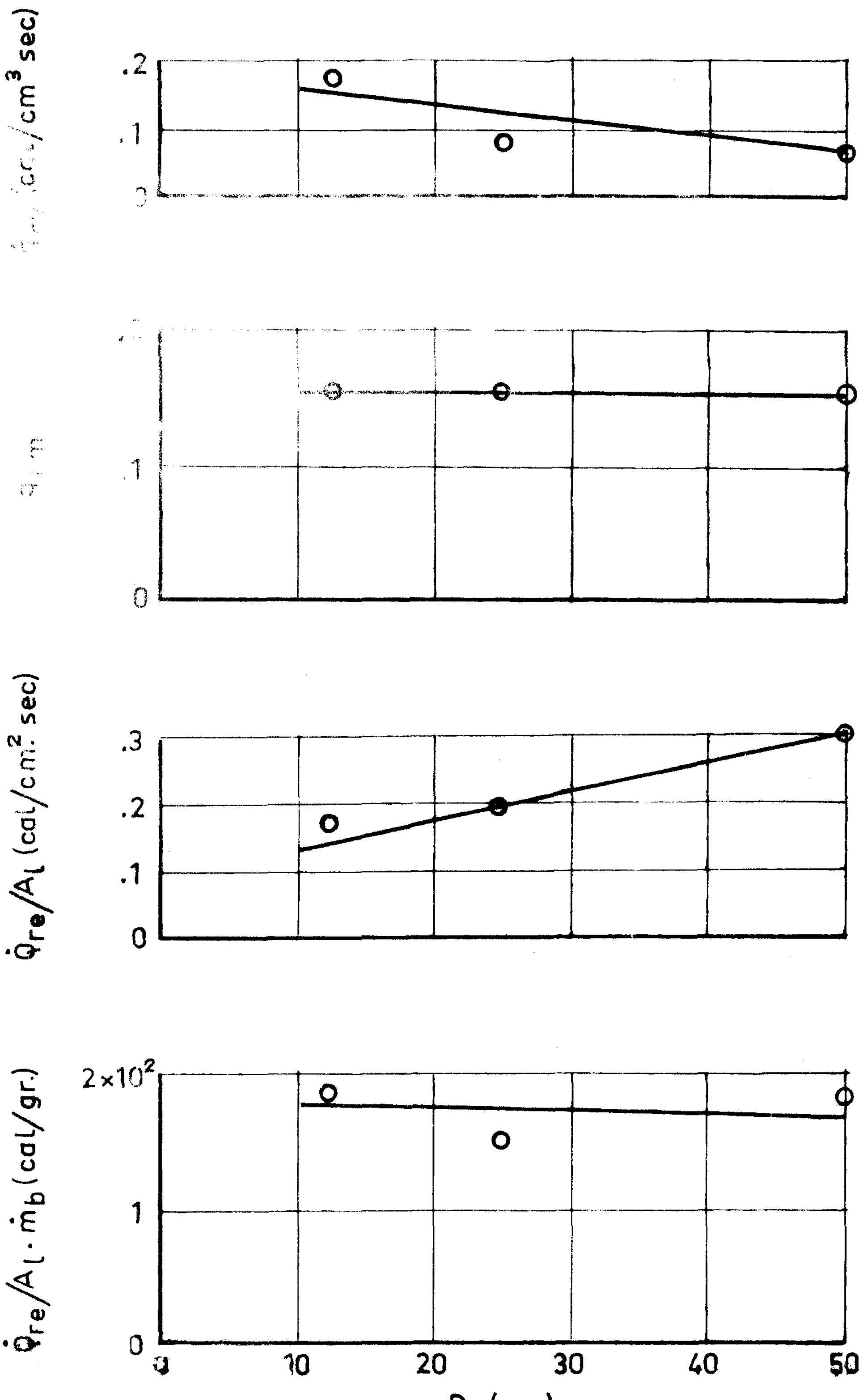


Fig.29. Influence of Diameter. Dioxane, ξ = 1 .

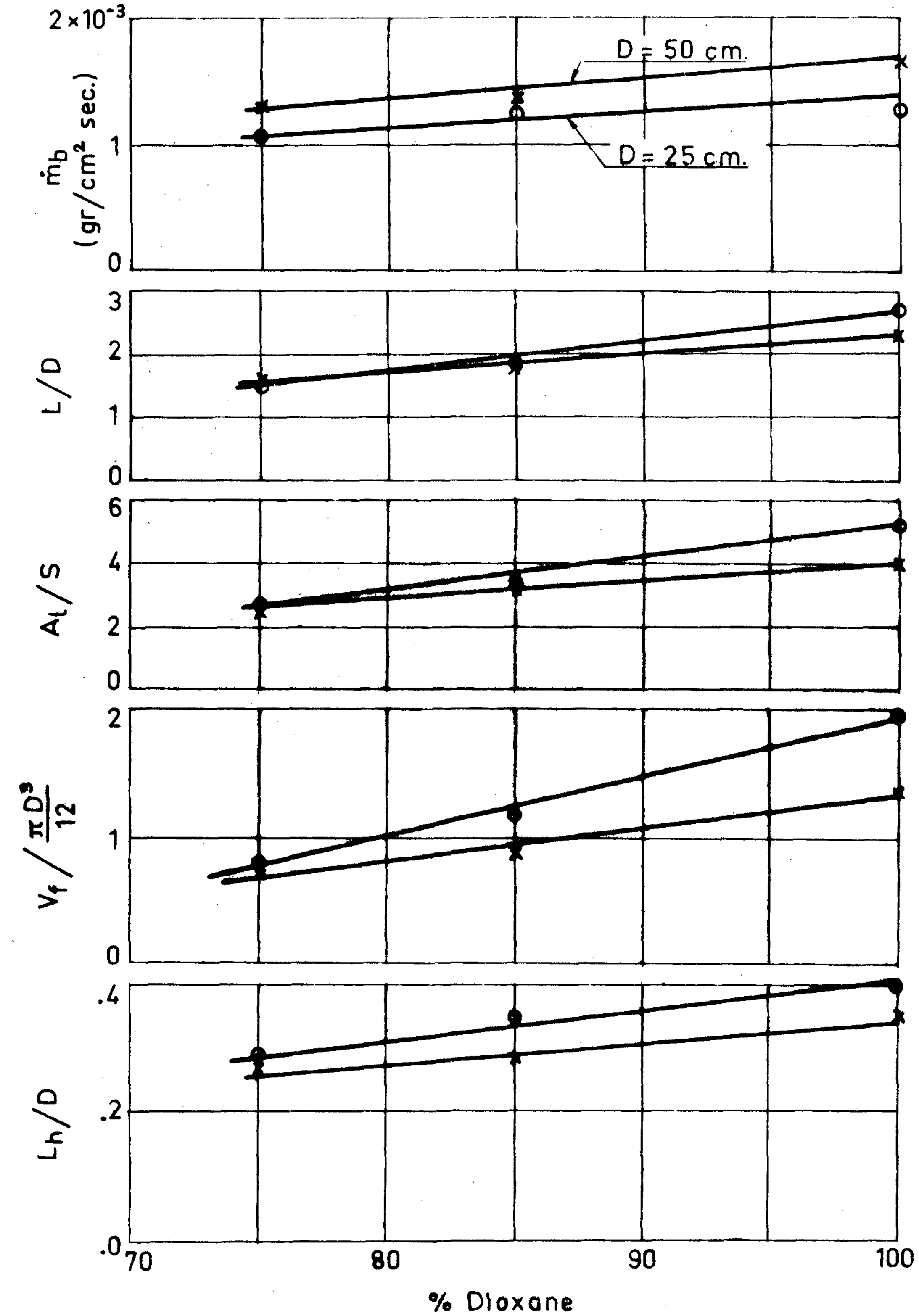


Fig.30. Influence of the Fuel Composition. Dioxane-Water Mixture.

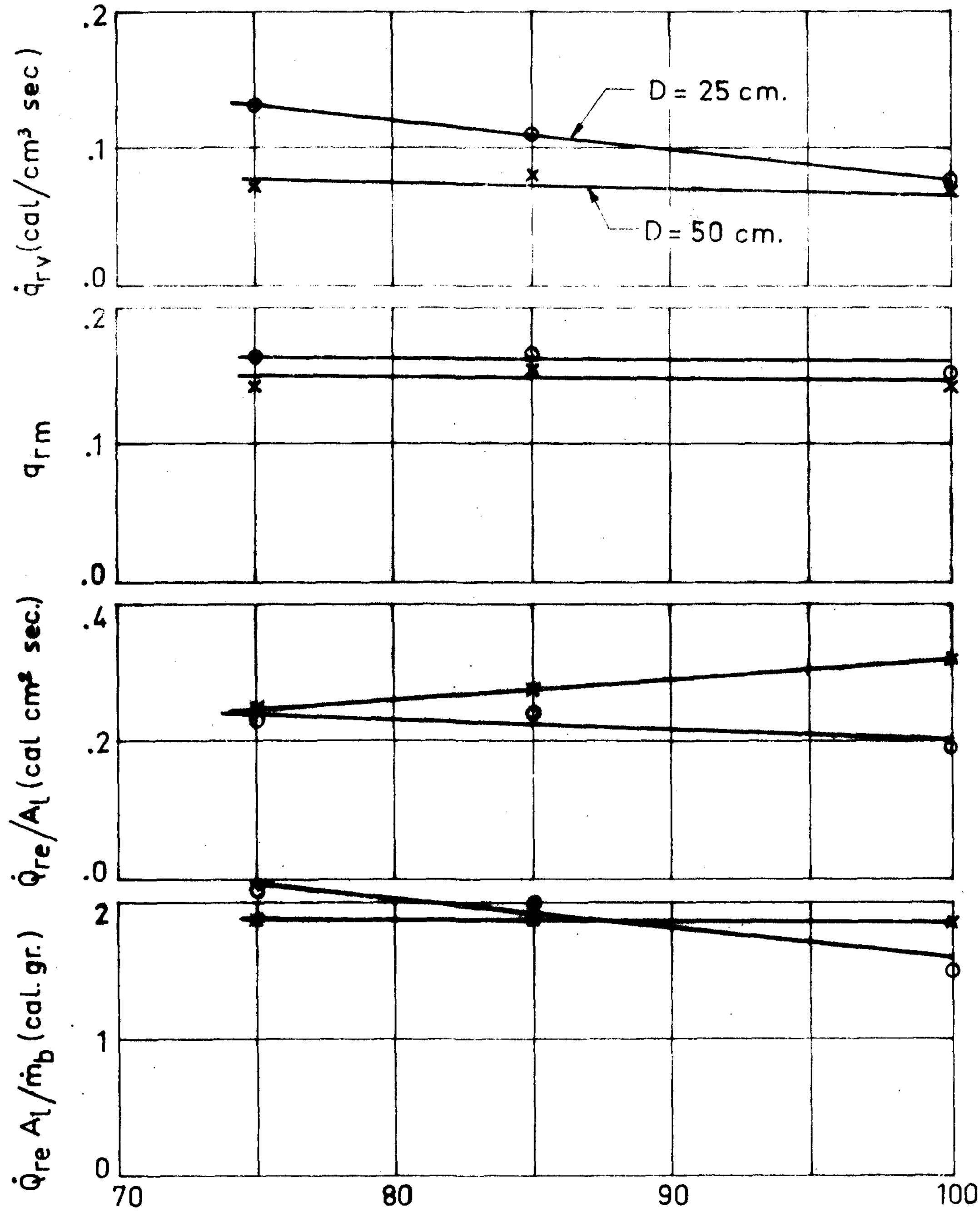


Fig.31. Influence of the Fuel Composition. Dioxane-Water Mixture.

flame size and flame shape present some scattering which is not observed when studying large flames.

3.4 Small vessels

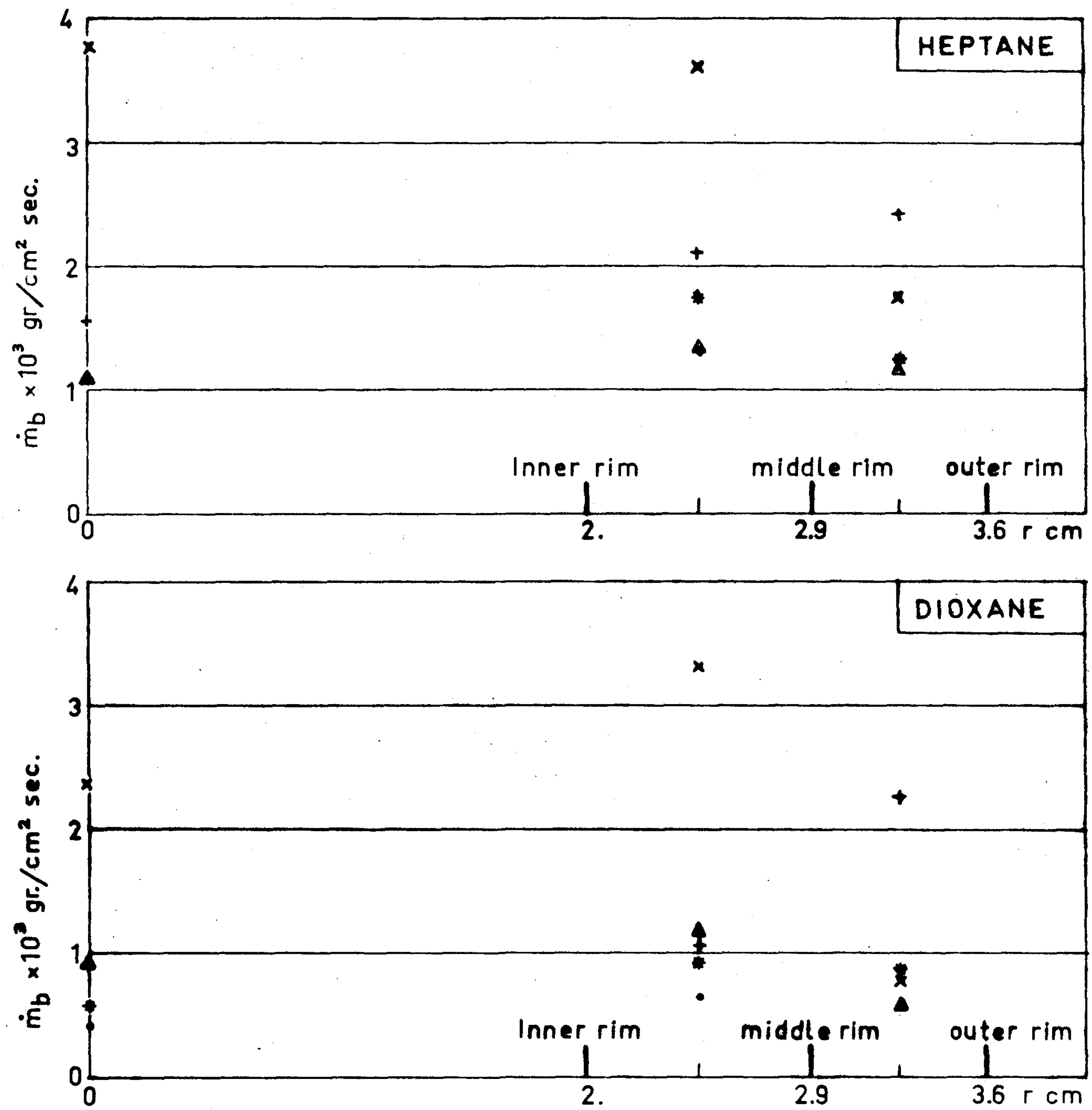
In Figs. 32 and 33 some experimental results obtained are shown. Each point represents the average value of at least ten measurements.

In Fig.32 it may be observed that $\dot{m}_b$ increases from the center of the vessel towards the rim. When vessel cooling is light results are in agreement with these predicted theoretically (Fig.11).

It may also be observed that burning rate decreases noticeably near the rim when vessel cooling is increased. Therefore, for small vessels, the convective heat flux from the vessel wall, either positive or negative, exerts an important influence on the value of the burning rate $\dot{m}_b$.

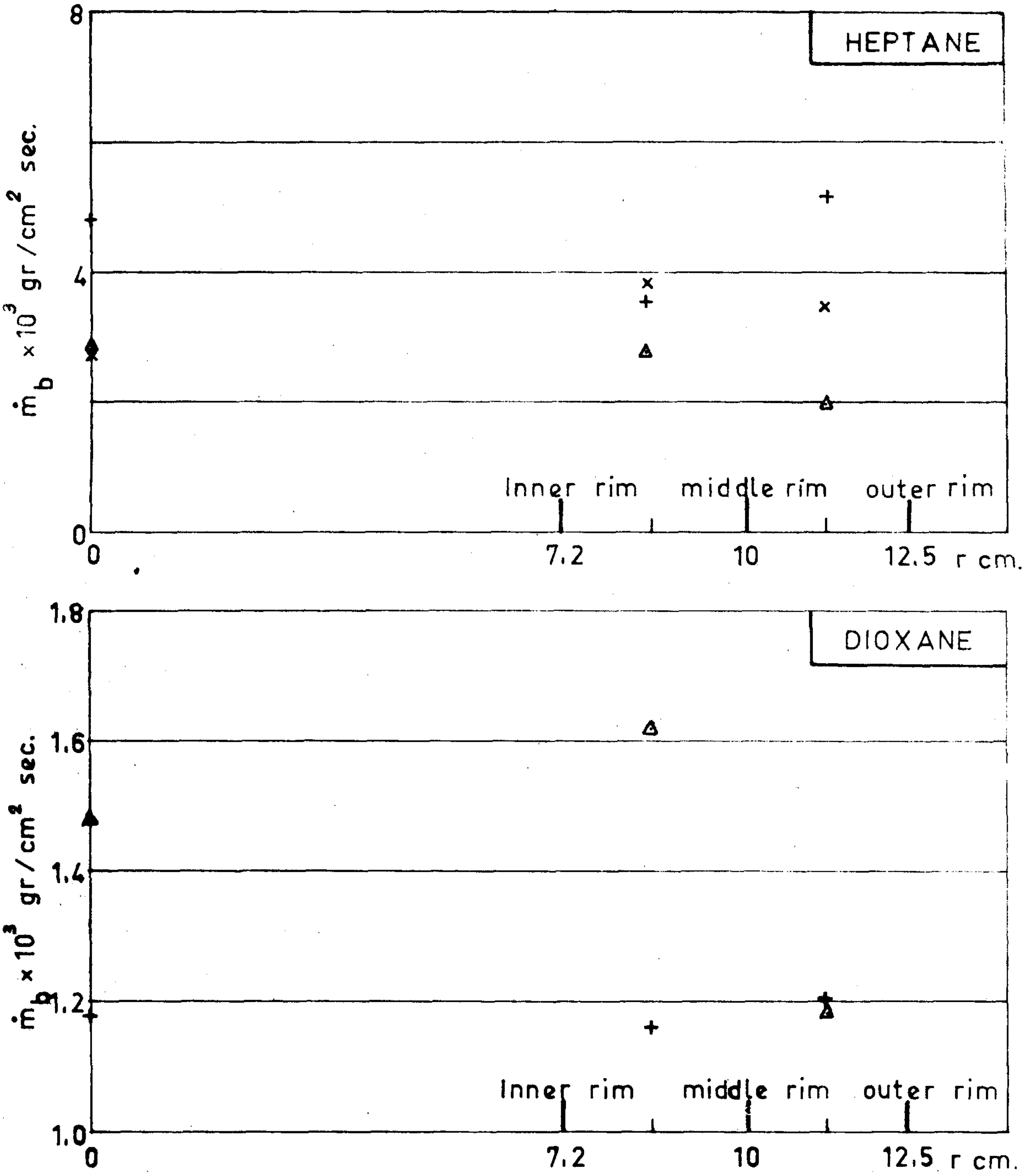
According to this, it may be concluded that the increasement of burning rate as vessel diameter is reduced, as stated by several investigators[4,5,6], depends, largely on the experimental technique.

When combustion takes place in only one of the annular compartment of the 72 mm vessels either at the central or at the middle one, combustion rate is higher. This is due to the heat transferred from the walls, since in this case this convection effect from the walls is more important than in the case of all compartments of the vessel being completely filled with fuel.



+ Combustion in all compartments without cooling
* " " " with "
× Combustion in one compartment, other compartments empty
▲ " " " " " " with water
• " " central and middle compartments with cooling

Fig.32. Radial Distribution of Burning Rate in Small Vessels.



+ Combustion in all compartments without cooling
× Combustion in one compartment without cooling
▵ Combustion in compartments with cooling

Fig.33. Radial Distribution of Burning Rate in Medium Sized Vessels.

When combustion takes place at the outer compartment combustion rate is lower because its outer wall is cooled.

For the large vessel (Fig.33) the distribution of $\dot{m}_b$ is similar when dioxane fuel is utilized. On the other hand, when n-heptane is burned the distribution of $\dot{m}_b$ changes and it is almost constant. This is due to the fact that the influence of radiation of the n-heptane is more important as compared with that of the dioxane flame.

III. DISCUSSION OF RESULTS AND CONCLUSIONS

1. INFLUENCE OF THE VESSEL DIAMETER

In Eq. (44), the influence of vessel diameter on the burning rate is exerted, essentially, through the radiation term, that is to say through the value of L_h .

It has been observed[4,5,6] as well as in our experimental results, that the law of variation of burning rate $\dot{m}_b$ as function of vessel diameter has a minimum value, which coincides with the transition of the flame from laminar to turbulent conditions.

From Figs.25, 27, 29 and 31 it is deduced that q_{rm} is constant regardless of the vessel diameter. The amount of heat received by the liquid fuel through radiation is:

$$S\dot{q}_{rf} = \phi_r \dot{Q}_r = \phi_r q_{rm} S\dot{m}_b q_r$$

For the case in which the convective heat $\dot{q}_{cv}$ is small, $\nu >> 1$, and $\xi = 1$, from Eqs. (16) and (43) it is obtained

$$\phi_r q_{rm} S\dot{m}_b q_r = S\dot{m}_b \left[q_l + c\,(T_s - T_o) \right]$$

that is

$$\phi_r = \frac{q_l + c\,(T_s - T_o)}{q_{rm} q_r} \qquad (58)$$

from which the conclusion is drawn that ϕ_r is constant regardless of the vessel diameter.

The following table shows the values of ϕ_r obtained from the above written expression and those calculated by assuming that the flame behaves as a black body.

	n-Heptane	Dioxane
$\phi_r = \dfrac{q_l + c_l\,(T_s - T_o)}{q_{rm} q_r}$	0,045	0,14
$\phi_r = \dfrac{\pi D^2}{\pi D^2 + 4A_l}$	0,06	0,18

It may be seen that a fair agreement between both results exists.

The value of L_h depends on both flame size and flame shape. Flame size depends on $\dot{m}_b$, and this burning rate as a first approximation is a function of q_r/q_l . As a consequence, the values of $L_h q_l/Dq_r$ have been calculated and are shown in Fig. 34 . It may be observed that they do not depend neither on vessel's diameter nor on the type of fuel. Their value are approximately given by:

$$\frac{L_h q_l}{D\,q_r} = 0.55 \,.\, 10^{-2}$$

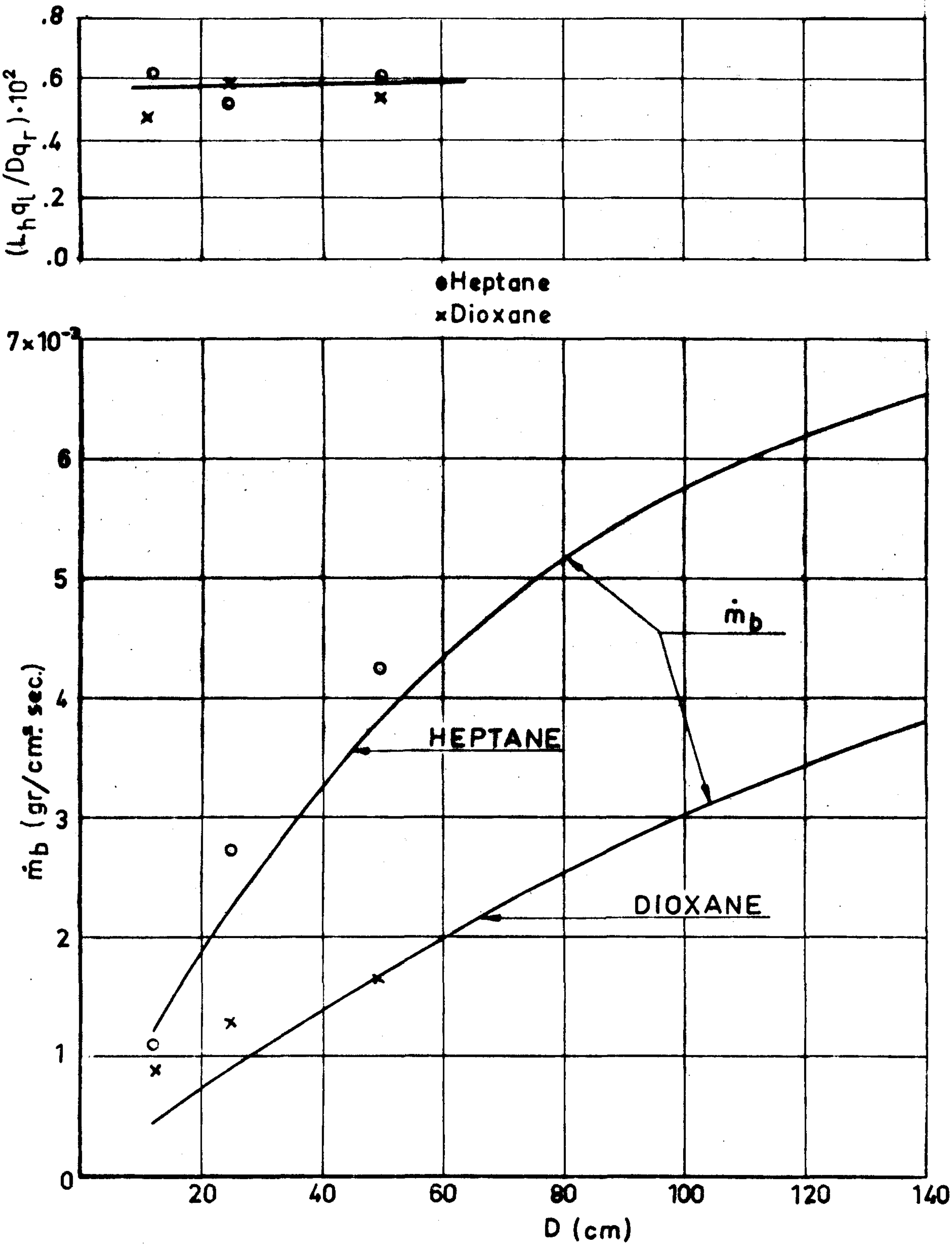


Fig.34. Influence of Diameter. Theoretical and Experimental Results.

The value of the absortion coefficient α given by Rasbash et al[1] is of the order of 0.02 (1/cm.)

From these values and disregarding convection $\dot{q}_{cv}$, from Eqs. (16) and (44) it is obtained:

$$\dot{m}_b = \frac{\sigma T_f^4}{q_1 + c\,(T_s - T_o)} \left[1 - \exp\left(0.55.10^{-2} \alpha \frac{q_r}{q_1} D\right)\right] \quad (59)$$

which gives the value of $\dot{m}_b$ as a function of both vessel diameter and radiant flame temperature.

The average value of T_f is of the order of 625°C and this temperature is similar for many types of fuels, as shown by the experimental data of Fons et al[3] as well as by our experimental results. With such values, Eq. (59) may be written in the form:

$$\dot{m}_b = \frac{0.9}{q_1 + c\,(T_s - T_o)} \left[1 - \exp\left(0.0055\, \alpha \frac{q_r}{q_1} D\right)\right] \quad (60)$$

in which $\alpha \simeq 0.02$ for flames with carbon particles in suspension (hydrocarbons) and $\alpha \simeq 0.015$ if such particles do not exist (alcohol flames or dioxane, for example).

Fig.34 also shows the values of $\dot{m}_b$ obtained from that expression as well as those experimentally obtained.

When the vessel diameter is large, Eq. (60) gives the same results as those given by the expression:

$$v_\infty = \frac{0.0076\ q_r}{q_1 + c\ (T_s - T_o)}\ \text{cm/min.}$$

derived by Grumer et al[7].

For very small values of the vessel diameter the influence of the walls (through convection) is important. Radiation does not control the phenomenon and Eq (60) no longer holds.

2. OVERFLOW INFLUENCE

The influence of the overflow, $\xi = \dot{m}_b/\dot{m}$, on the burning rate is given by Eq.(44) taking acount the values of ν_v and ν.

When $\nu >> 1$, which is the normal case, Eq. (44) reduces to:

$$\dot{m}_b = \frac{\varepsilon_f \sigma T_s^4 \theta_f^4 + \dfrac{T_s}{z_{v,min}} (\theta_f - 1)\ \phi_v}{q_1 + T_s\ (1 - \theta_o)\ \dfrac{c}{\xi}} \qquad (61)$$

This equation gives, within a first-orden approximation, the influence of the overflow ξ on burning rate.

It may be seen that the larger is the specific heat c and the smaller the heat of vaporization q_1 , the greater is the influence of the overflow.

This influence of ξ is important because the spilled fuel (overflow) absorbs an important percentage of the heat received by the fuel, which reduces $\dot{m}_b$ and the flame size, and in turn, the heat transmitted to the fuel.

In order to compare the theoretical and experimental results and in order to isolate the influence of ξ, the value of $\dot{q}_s/\dot{m}_b = q_1 + T_s\ (1 - \theta_o)\ c/\xi$, obtained from Eq.(61) and the experimental value of $\dot{Q}_{re}/A_1\dot{m}_b = \dot{q}_s/\dot{m}_b$ are compared in Fig.35. Both values are in fairly good agreement. The constant separation between both curves may be due to the heat lost by the fuel through the vessel walls. When these two values are compared, the effect produced by the decreasement of the flame size due to the reduction of the value of ξ is eliminated. Therefore, heating of the spilled fuel (overflow) is the only effect compared.

3. INFLUENCE OF THE TYPE OF FUEL

The influence of the type of fuel has been studied by comparing the results obtained by burning n-heptane, pure dioxane, and mixtures of dioxane and water.

The two more important properties of fuels as far as open fires is concerned, are the heat of vaporization q_1 and heat of combustion q_r.

It seems that the influence of fuel composition on the burning rate depends on vessel size, because the relative importance of the two heat transfer mechanism: radiation and convection, depends on this vessel size.

For large vessels, the value of q_r/q_1 influences the flame size and then, its emissivity. In Fig.34 it may be

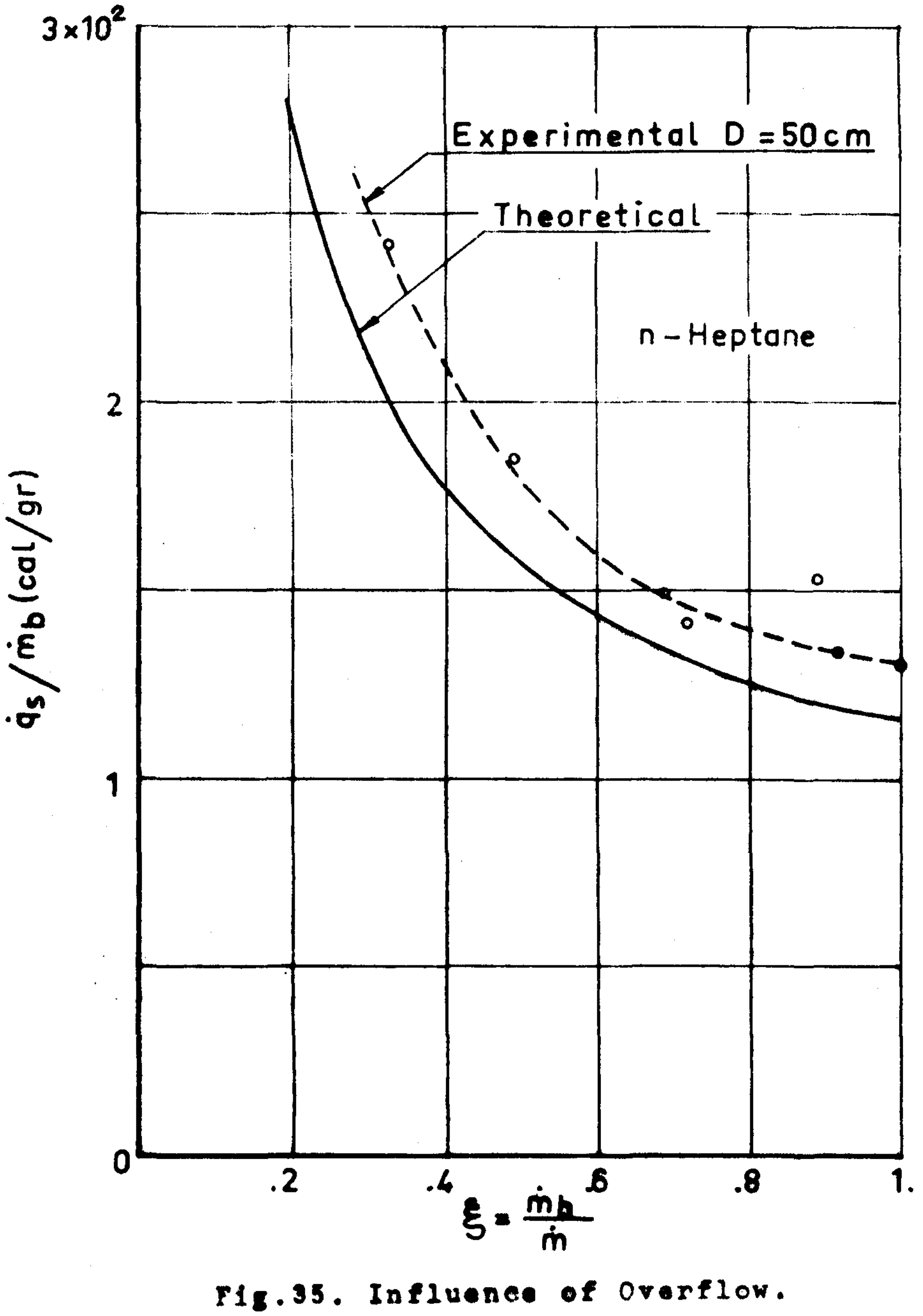


Fig.35. Influence of Overflow.
Theoretical and Experimental Results.

observed that the influence of fuel composition is more important as the vessel diameter increases.

For small vessels, in which the relative importance of the radiation heat transfer is small, the influence of fuel composition is less important, resulting similar values of $\dot{m}_b$ for n-heptane and dioxane. This is probably due to the fact that an important percentage of the heat transferred to the fuel is through the vessels walls, and their heat transfer depends, largely, on the local flame temperature.

4. GENERAL EXPRESSION OF THE BURNING RATE

Considering that the influence of radiation is not important for small vessel diameter and that, on the other hand, convection is negligeable for large diameters, both effects can be superimposed. Then, Eqs. (50), (57), and (59) give the following expression of the burning rate, valid for all range of diameters:

$$\dot{m}_b = \frac{\lambda_v}{c_v z_{v,min}} \; \frac{2(\beta \ln\beta - \beta + 1)}{(\beta - 1)^2} \ln\left[1 + \frac{c_v\,(T_f - T_s)}{q_l + c(T_s - T_o)/\xi}\right] +$$

$$+ \left[\frac{7{,}44\,\left(\frac{c}{\lambda}\right)^{1/3} D^{-4/3}\,\Delta T}{q_l + c\,(T_s - T_o)/\xi}\right]^{3/2} +$$

$$+ \frac{\sigma T_f^4}{q_l + c_l\,(T_s - T_o)/\xi}\left[1 - \exp\left(-0{,}0055\alpha\,\frac{q_r}{q_l}\,D\right)\right] \quad (62)$$

in which

$$\Delta T = T_w - \frac{T_s + T_o}{2} \quad , \quad \beta = \frac{D}{2\ z_{v,min}} \quad , \quad T_f \simeq 650°C .$$

In Fig. 36, results obtained from this expression are represented.

5. TRANSIENT PROCESS

The Eq. (45) for the normal case in which $\nu >> 1$, may be written in the form:

$$\frac{\dot{m}_b}{\dot{m}_{b\infty}} = 1 - \frac{(T_s - T_o)\lambda}{\dot{m}_{b\infty} q_1\ x_s} \sum_n \frac{8\pi^2 n^2}{\nu^2 + 4\pi^2 n^2} \exp\left[\frac{(-4\pi^2 n^2 + \nu^2)\tau}{4}\right] +$$

$$+ \frac{(T_f - T_s)\ \lambda_v}{\dot{m}_{b\infty} q_1\ z_{v,min}} (\phi_v - \phi_{v\infty}) - \frac{\dot{q}_{r\infty} - \dot{q}_r}{\dot{m}_{b\infty}\ q_1} \tag{63}$$

in which subscript ∞ indicates stationary conditions.

The two last terms of this equation decrease as time increases. ϕ_v decreases and tends towards $\phi_{v\infty}$ and the value $\dot{q}_r$ augments.

Fig. 37 shows the experimental results for a typical case, as well as the value of

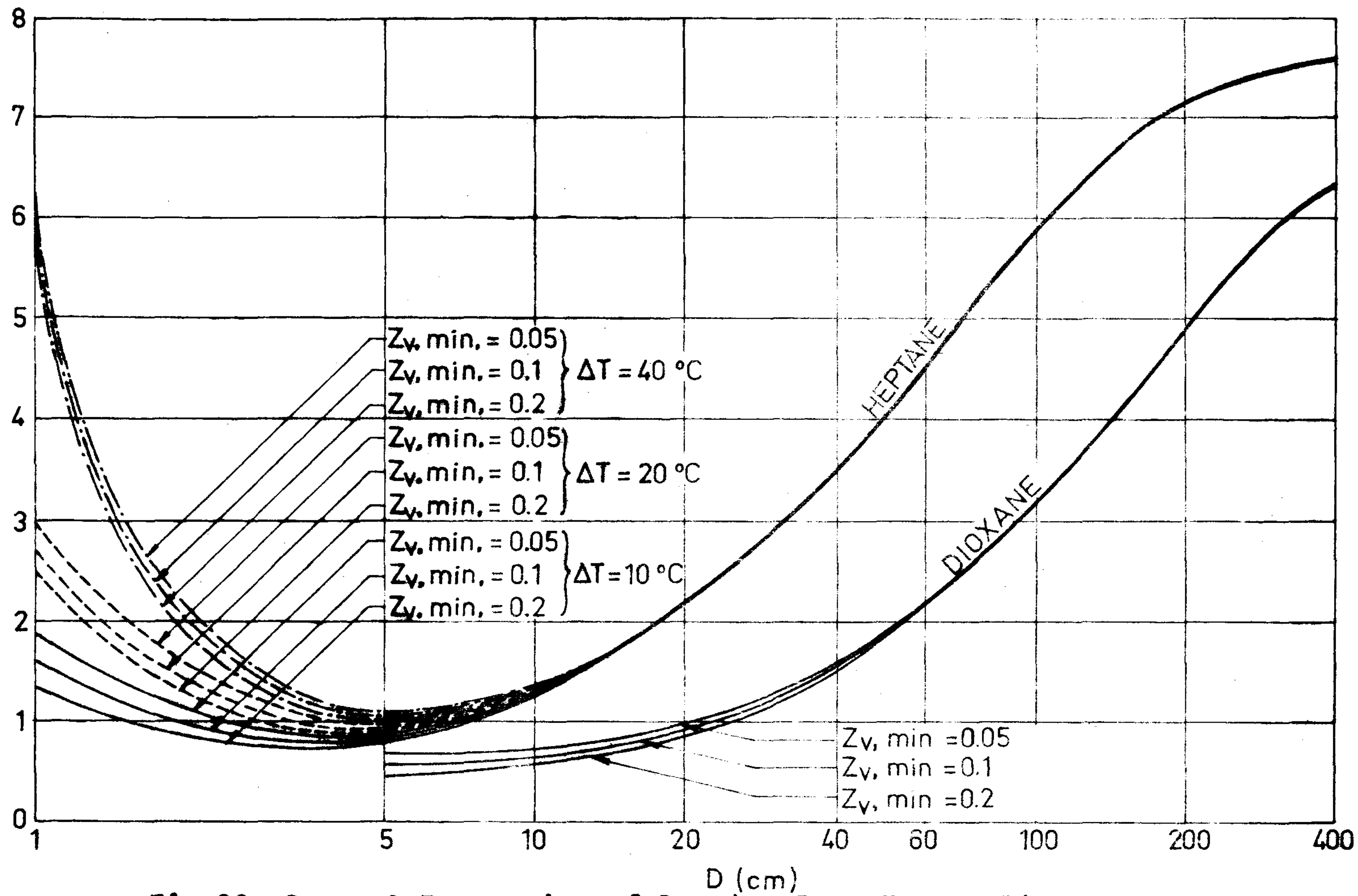


Fig.36. General Expression of Burning Rate Versus Diameter.

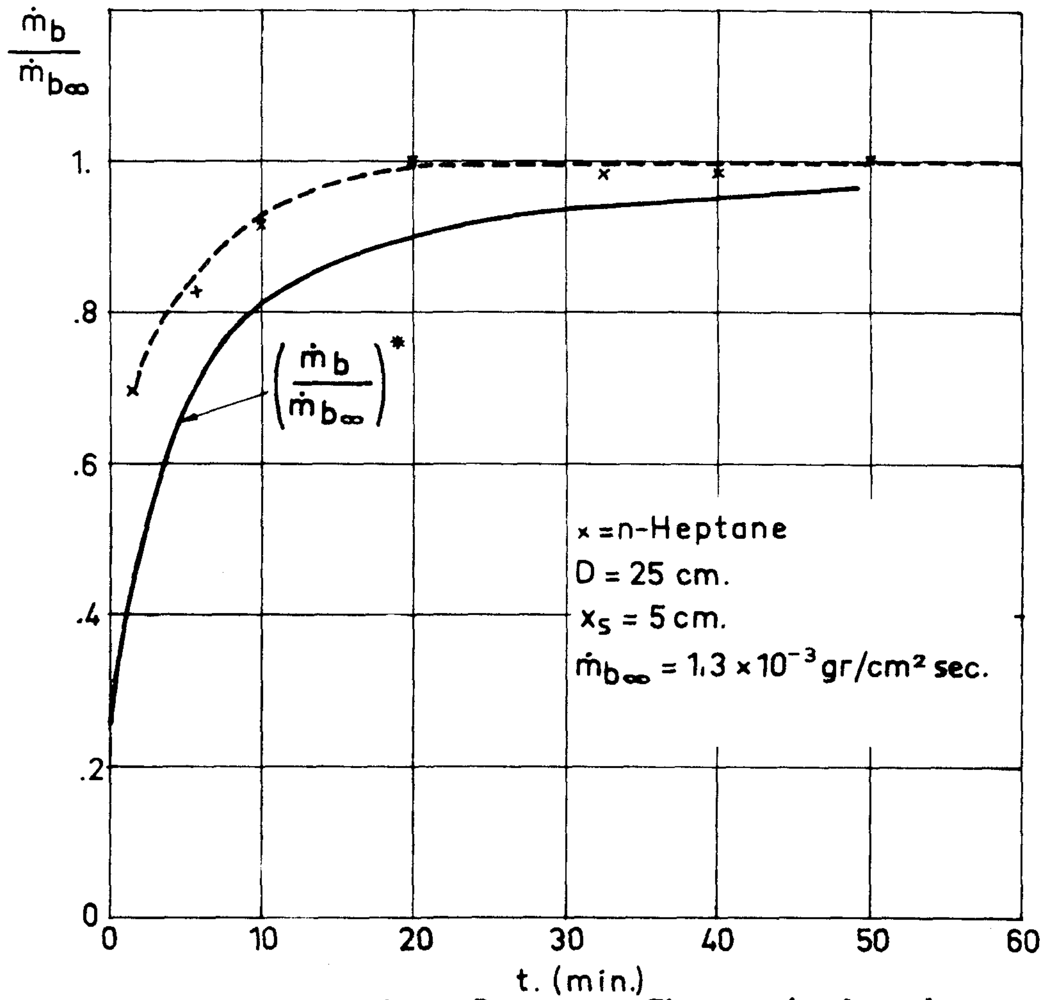


Fig.37. Transient Process. Theoretical and Experimental Results.

$$\left(\frac{\dot{m}_b}{\dot{m}_{b\infty}}\right)^{*} = 1 - \frac{(T_s - T_o)}{\dot{m}_{b\infty}\; q_1} \frac{\lambda}{x_s} \sum_n \frac{8\ \pi^2 n^2}{\nu^2 + 4\pi^2 n^2} \exp\left[- \frac{(4\pi^2 n^2 + \nu^2)\tau}{4}\right]$$

The separation between the two curves is due to the existence of the aforementioned last two terms of Eq. (63).

6. CONCLUSIONS

The principal conclusions obtained are the following:

1º- In a pool fire temperature profiles in the liquid fuel are very sharp in the vicinity of the fuel free surface. Therefore,a thermocouple placed at the fuel surface would measure an average value of the temperature, somewhat smaller than its superficial value, which should be very close to the boiling temperature at ambient pressure.

2º- Liquid depth influences considerably the time required to achieve quasi-stationary conditions, and this time may be extremely long.

3º- The influence of the overflow on the burning rate is important. The value of $\dot{m}_b$ may be reduced by a factor higher than two.

4º- The value of the overflow does influence all flame properties (size, shape, radiation, etc).

5°.- The heat received through convection by the fuel decreases rapidly as $\dot{m}_b$ augments.

6°.- Total energy radiated by the flame per unit value of $\dot{m}_b$ does not depend on vessel diameter.

7°.- The influence of the fuel properties on the burning rate depends on vessel diameters. For large diameter that influence is more noticeable.

8°.- By mixing dioxane with water it is possible to change continuosly the flame properties, specially the heat of reaction and the latent heat of evaporation.

9°.- An analytical expression giving burning rate distribution as function of vessel radius has been obtained.

10°.- An analytical expression for the over-all burning rate as function of the physical properties of fuel (c_v , λ , q_l , T_s , q_r), as well as function of both flame radiant properties (α, T_f) and vessel diameter has been derived.

11°.- The influence of the heat transmitted through the lateral walls of vessel on the burning rate has been expressed analytically.

12°.- For small vessel it has been found that the influence of the heat transmitted through the lateral walls on the burning rate is very important.

NOTATION

A_{fr}	Cross-section area of flame
A_{ft}	Total area of flame
A_l	Lateral area of flame
c	Specific heat of the liquid fuel
c_v	Specific heat of the fuel vapor
D	Burner diameter, Fig.1
h	Convective heat transfer coefficient
L	Flame height, Fig.1
L_h	Mean beam length
$\dot{m}$	Fuel flow within the vessel
$\dot{m}_b$	Burned fuel flow
$\dot{m}_{b'}$	Burned local fuel flow
$\dot{m}_{b'o}$	Local burning rate at the center of the vessel
N_u	Nusselt's number
$\dot{q}_{cl}$	Heat transferred by conduction throughout the liquid per unit area and per unit time
$\dot{q}_{cv}$	Heat transferred by convection from the flame to the liquid surface per unit area and per unit time
q_l	Latent heat of evaporation
q_r	Heat of reaction
$\dot{Q}_r$	Total heat radiated per unit time
$\dot{Q}_{re}$	Heat radiated to the surroundings per unit time
$\dot{Q}_{rf}$	Total heat radiated to the fuel per unit time
$\dot{q}_{rf}$	Heat radiated to the fuel per unit area and per unit time
$\dot{q}_{rv}$	Total heat radiated per unit volume of flame
q_{rm}	Total heat radiated per unit of burning rate
$\dot{q}_s$	Heat received by the fuel per unit area and per unit time

$\dot{Q}_w$	Heat transferred from the wall
r	Burner radius
S	Burner surface
t	Time
T_f	Flame temperature
T_o	Ambient temperature
T_m	Average temperature
T_s	Surface temperature
v	Liquid fuel velocity
V_f	Flame volume
x	Vessel (fuel) depth. Fig.1
y	Flame horizontal coordinate, Fig.1
z_v	Vertical coordinate in the vapor zone, Fig.1
α	Emission coefficient
β	Dimensionless parameter. Eq. (28)
Γ	Dimensionless parameter. Eq. (19)
δ	Dimensionless parameter. Eq. (4)
δ_v	Dimensionless parameter. Eq. (22)
ε_f	Flame emissivity
ξ	Overflow ratio defined by $\dot{m}_b/\dot{m}$
λ	Thermal conductivity
ϕ_v	Dimensionless parameter defined by Eq. (30)
ϕ_r	Dimensionless parameter defined by $\dot{Q}_{rf}/\dot{Q}_r$
κ_1	Dimensionless parameter, Eq. (7)
κ_s	Dimensionless parameter, Eq. (6)
ν	Dimensionless parameter, Eq. (8)
ν_v	Dimensionless parameter, Eq. (21)
ρ	Fuel density
σ	Stefan-Boltzmann constant
τ	Dimensionless parameter, Eq. (5)

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